THE GAS, PETROL, AND OIL ENGINE

THE GAS
PETROL, AND OIL ENGINE

VOL. I.

THERMODYNAMICS OF THE GAS, PETROL, AND OIL ENGINE, TOGETHER WITH HISTORICAL SKETCH

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PREFACE

THIS work was first published in 1886 under the title of 'The Gas Engine.' Considerable additions were made to it in 1896 and the title was altered to 'The Gas and Oil Engine.' Both the science and practice of Gas, Petrol, and Oil Engines have developed so largely since 1896 that it has become necessary to rewrite practically the whole book. To treat science and practice adequately necessitated considerable additional space; and, accordingly, the book has been divided into two volumes, each, however, complete in itself. The first volume has been entitled 'Thermodynamics of the Gas, Petrol, and Oil Engine,' and the second volume will be entitled 'The Gas, Petrol, and Oil Engine in Practice.' The present volume consists of an enlarged historical sketch, broadly dealing with the important developments down to 1908, and the science of the subject is treated in nine chapters. In this treatment the original book is followed closely, so far as arrangement is concerned. Chapters I. and II. deal with the gas engine method and classification of gas engines. Chapter III. on 'Thermodynamics' has been greatly enlarged to deal more fully with cycles of operation of practical importance to-day. Chapter IV., on 'The Causes of Loss in Gas Engines,' has also been enlarged. Chapter V., on 'Combustion and Explosion,' has been added to. Chapter VI. has been greatly enlarged to include the consideration of Cooling as well as Explosion in a Closed Vessel and the more recent experiments of the Massachusetts Institute of Technology, Grover, Clerk, the Royal College of Science, Petavel, and Hopkinson have been fully dealt with. Information upon the Laws of Explosion and Cooling has greatly increased even within the last six years. The work of the various investigators has been described in a manner which it is hoped will be useful to the engineer. Chapter VII. deals with the Discussion of Data obtainable from the work of various experimenters as to the Laws of

Explosion and Cooling of Gaseous mixtures in large and small vessels and at initial pressures of atmosphere and above. Chapter VIII. deals for the first time in any work on the subject with Explosion and Cooling in a Cylinder behind a Moving Piston. The recent work by the author enables approximate values to be arrived at for cooling within the Internal-Combustion Engine Cylinder, apart from temperature fall due to work done. The last chapter, Chapter IX., discusses the Thermal and Mechanical Efficiency of all the different types of gas engine in use. On this part of the subject more accurate knowledge exists than at any previous time. Important work has been done by English, American, and Continental investigators, and the Research Committees of the Institutions of Civil and Mechanical Engineers have made experiments of great value. All this work has been fully discussed in this chapter.

In the present volume the author has attempted to systematise the knowledge existing as to the properties of the working fluid of the Internal-Combustion Engine, whether using gas, petrol, or heavy oil, so as to enable the engineer and inventor to consider not only mechanical modifications of engine construction, but more profound alterations possible by varying the actions going on in the working fluid. Such variations are, in the author's opinion, necessary to enable light and powerful Internal-Combustion Engines to be developed for marine work. For this purpose it is necessary that the engineer should be thoroughly familiar with the properties of the working fluid with which he is dealing.

Appendices have been added to make clear many of the properties of gaseous explosions, and the valuable Report of the British Association Committee on Gaseous Explosions has been published in full in Appendix IV. This Report, published at Section G, Dublin, in 1908, contains the latest information available on all the properties of gaseous explosions.

The author has freely availed himself of the various important Papers and Reports published by the Royal Society, the Institution of Civil Engineers, and the Institution of Mechanical Engineers. He is greatly indebted to the Institution of Civil Engineers for permission to use many blocks, and is much indebted also to the Institution of Mechanical Engineers for permission to reproduce figs. 108 to 118 inclusive.

The author has much pleasure in thanking his assistant, Mr. W. Grylls Adams, M.A., for his effective aid in preparing and checking the numerous calculations and curves which are given, as well as reading the proofs.

He is also pleased to thank his assistant, Mr. Aubrey T. Evans, for the preparation of the Index.

Engineering Laboratory, 6 Featherstone Buildings,
High Holborn, London. *June* 1909.

CONTENTS

APPENDICES

LIST OF PLATES

THERMODYNAMICS

OF THE

GAS, PETROL, AND OIL ENGINE

HISTORICAL SKETCH OF THE GAS, PETROL, AND OIL ENGINE

THE origin of the gas engine is but imperfectly known; by some it is dated as far back as 1680, when Huyghens proposed to use gunpowder for obtaining motive power. Papin, in 1690, continued Huyghens' experiments, but without success. The method used was a fairly practicable one. The explosion was used indirectly; a small quantity of gunpowder exploded in a large cylindrical vessel filled with air expelled the air through check valves, thus leaving, after cooling, a partial vacuum. The pressure of the atmosphere then drove a piston down to the bottom of the vessel, lifting a weight or doing other work.

In a paper, published at Leipsic in 1688, Papin stated that 'until now all experiments have been unsuccessful; and after the combustion of the exploded powder, there always remains in the cylinder about one-fifth of its volume of air.'

The Abbé Hautefeuille made similar proposals, but does not seem to have made actual experiments. These early engines cannot be classed as gas engines. The explosion of gunpowder is so different in its nature from that of a gaseous mixture that comparison is untenable. The first real gas engine described in this country is in Robert Street's patent, No. 1983, 1794. It contains a motor cylinder in which works a piston connected to a lever, from which lever a pump is driven. The bottom of the motor cylinder is heated by a fire; a few drops of spirits of turpentine being introduced and evaporated by the heat, the motor piston is drawn up, and air entering mixes with the inflammable vapour, the application of a flame to a touch-hole causing explosion; and the piston being driven up forces the pump piston down, so performing work in raising water. The details

as described are crude, but the main idea is correct and was not improved upon in practice till very lately.

Lebon, in France, describes a gas engine in his French patent, No. 356 of September 28, 1799. In it gas and air are supplied from separate compressing pumps to a combustion chamber where the gases are detonated. A motor cylinder is supplied from this chamber with the hot gases under pressure by distributing valves contained in a valve box. Both motor and pump cylinders are double-acting. The engine resembles what became known later as a constant-pressure engine, but the inventor's notions were vague, and he does not distinguish very clearly between explosion and constant pressure. He expects, however, to greatly improve upon the steam engine, as he states: 'But experiments on this force teach that the height should be prodigiously superior to that which measures the force of our fire engines.' The engine shown in the patent drawings is very crude; it could not have been in practical operation.

Poor Lebon had but little time to develop his ideas, as he was assassinated in 1804.

In the year 1820 the Rev. W. Cecil, M.A., of Cambridge, read a paper at the Cambridge Philosophical Society with the following title: 'On the Application of Hydrogen Gas to Produce a Moving Power in Machinery, with a description of an Engine which is moved by the Pressure of the Atmosphere upon a Vacuum caused by Explosions of Hydrogen Gas and Atmospheric Air.' In this paper he described an engine which he had constructed to operate according to the explosion vacuum method; and he stated that at sixty revolutions per minute the explosions take place with perfect regularity. His engine consumed, he stated, 17·6 cubic feet of hydrogen gas per hour. His hydrogen explosion appears to have been accompanied by considerable noise, because he states with regard to a proposed larger engine, '. . . to remedy the noise which is occasioned by the explosion, the lower end of the cylinder A, B, C, D may be buried in a well, or it may be enclosed in a large air-tight vessel.' In this paper he also mentions an engine operated by non-compression explosion and also one operated by gunpowder. This paper gives an account of the first gas engine which appears to have been worked in Britain, and, it is believed, in the world.

Cecil appears to have been the first to attempt to measure the pressures produced by gaseous explosions; for this purpose he used a tin cylinder, ten inches long by two inches diameter, 'made of thin tin, seamed up one side, and soft-soldered, the ends being well secured. This vessel, he states, will easily sustain without bursting the whole force of an exploding mixture of hydrogen and air: this force he deter-

mines as 180 lb. per square inch absolute, as found in the following manner.

'The greatest expansive force was ascertained by filling with mixed gas the cylinder just described, one end being entirely solid, the other being closed with a cork bung accurately fitted, and confined by several strings, parallel to the axis of the cylinder, and so arranged that the tension might be equally distributed. It was observed how many strings the explosion was able to break by pressing on a surface of three square inches. The same strings were then transferred to a common steelyard, and it was observed how much weight they would sustain. The result of several trials, differing but little from each other, indicated a pressure of 500 lbs. upon the three square inches. If to this be added 45 lbs. for the atmospheric pressure on the same surface, the whole being divided by 3 gives 180 lbs. nearly for the pressure upon every square inch.

Cecil's result was much too high, but his method of procedure was interesting and ingenious; his paper is important and shows very considerable knowledge of the problem to be solved. The engine actuated by 'the exploding force of the mixed gas' alluded to in the paper is stated to have been exhibited in operation 'about three years ago'—that is, in 1817—at the philosophical lectures of Professor Farish.

Samuel Brown's inventions come next. His patents are dated 1823 and 1826, Nos. 4874 and 5350. The principle used is ingenious and easily carried out in practice, but it is not economical, and it gives a very cumbrous machine for the amount of power produced. A partial vacuum is produced by filling a vessel with flame and expelling the air it contains; a jet of water is thrown in and condenses the flame, giving vacuum. The atmospheric pressure thus made available for power is utilised in any engine of ordinary construction.

Brown's apparatus consists essentially of a large upright cylindrical vessel fitted on the top with a movable valve cover, of the whole diameter of the cylinder. The cover is raised and lowered from and to its seat by a lever and suitable gear at proper times. The gas supply pipe enters the cylinder at the bottom; the cylinder being filled with air, and the valve raised, the gas cock is opened and the issuing gas lighted by a small flame as it enters the cylinder. The flame produced fills the whole vessel, expelling the air it contains; the valve being now lowered and the gas supply shut off, the water-jet is thrown in and causes condensation. To keep up a constant supply of power several of these cylinders are required, so that one at least may be always vacuous while the others are in the process of obtaining the vacuum. In the specification three are shown and three engines. The engines are all connected to the same crank-shaft. Notwithstanding this provision the motion must have been

irregular. The idea was evidently suggested by the condensing steam engine ; instead of using steam to obtain a vacuum flame is employed. Brown's engine, although uninteresting theoretically, is important as being the first gas engine undoubtedly commercially at work.

Brown was able and persevering, and appears to have been a man of business as well as an inventor : he succeeded in forming commercial companies for working his engines as applied to three purposes, pumping for water works, road locomotion and boat propulsion.

According to the 'Mechanics' Magazine,' published in London in August 1824, a model had been already made which raised 300 gallons

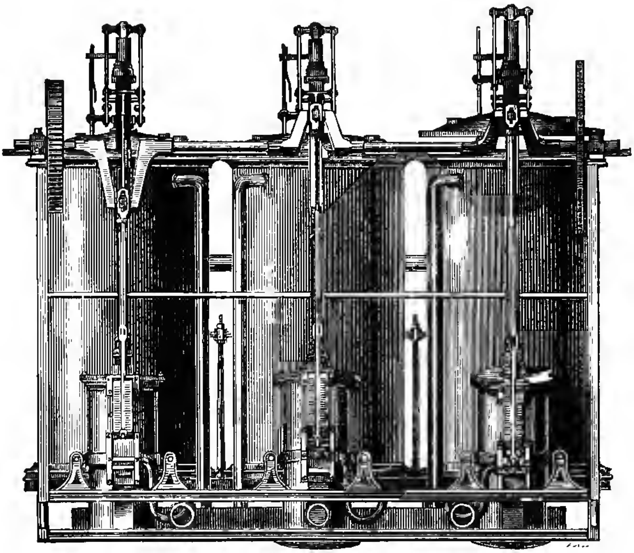


FIG. 1.—Brown's Gas-vacuum Engine, 1826

of water 15 feet high on one cubic foot of gas. In 1832 it appears four engines were in use for pumping :

(1) One at Croydon on the canal, raising water from a lower to a higher level.

(2) One at Soham in Cambridgeshire, for draining part of the middle Fen district.

(3) One at Eagle Lodge, Old Brompton.

(4) One at Eagle Lodge, Old Brompton, but of the beam type.

It is stated that the cylinder of the Croydon engine was 3 ft. 6 in. diameter by 22 in. high. Engine No. 3 above was inspected by the Editor of the 'Mechanics' Magazine' at work : its cylinder was 3 ft. 8¾ in.

diameter by 22 in. high ; and it discharged 750 gallons per stroke, four strokes per minute, 12 feet high.

Brown claimed, in a circular published in 1832, that the coke and tar obtained in making coal gas for the Croydon engine was sold for such sums as produced a profit in addition to giving motive power for nothing. He states that the whole annual expense of the Croydon gas vacuum engine, including coal, wages, repairs, depreciation, and rent, amounted to 666*l*. 14*s*., while the receipts from the sale of coke and tar were 769*l*. 12*s*., so that the annual profit was 102*l*. 18*s*., without counting the value of the work done, which previously cost the canal company 275*l*. per annum to effect by steam engine.

This state of affairs could not have lasted, as after some years' work the engines were dispensed with.

In 1825 Mr. J. A. Whitfield, of Bedlington Ironworks, describes Brown's engine as applied to a carriage, with sections of the cylinders, and a drawing of the carriage to scale. The wheels were 5 feet in diameter, wheel base was 6 ft. 3 in., and track 4 ft. 6 in. ; the weight, with gas and water, was 20 cwts. The cylinders were 12 in. diameter by 24 in. stroke.

It was later stated that this carriage successfully ascended the steepest part of Shooter's Hill, where the gradient was 13½ in. in 12 feet ; the ascent was made with considerable ease. The date of this test was the last week of May 1826.

In January 1827 a boat was propelled by Brown's engine on two days—the 1st and the 31st. In the first test the boat leaked (as the result of a collision the day before) and the river was rough, so that the trial was unsatisfactory, although the boat made some headway.

On January 31 everything went well, and, starting from Blackfriars Bridge, the boat travelled on the Thames at the rate of seven to eight miles per hour, it is said, 'with all the regularity of a steamer, and the paddles worked quite smoothly, and seemed capable of continuing to go as long as gas was supplied.'

The boat was 36 feet long, and the weight of the engine and framework came to 600 lbs. Mr. Brown stated that this test was made in the presence of the Lords of the Admiralty and a number of scientific men. Notwithstanding this partial success, the company formed to apply the engine to vessels dissolved in February 1827.

Samuel Brown deserves the greatest credit for his able and persevering attempts to introduce gas power for the purpose of locomotion on land and water, and he appears to be the first to make even an experimental application in a practicable form on a considerable scale.

W. L. Wright, 1833, *No*. 6525.—In this specification the drawings are very complete and the details are carefully worked out. The explosion of a mixture of inflammable gas and air acts directly upon

the piston, which acts through a connecting rod upon a crank-shaft. The engine is double-acting, the piston receiving two impulses for every revolution of the crank-shaft. In appearance it resembles a high-pressure steam engine of the kind known as the table pattern. The gas and air are supplied to the motor cylinder from separate pumps through two reservoirs at a pressure a few pounds above atmosphere ; the gases (gas and air) enter spherical spaces at the ends of the motor cylinder, partly displacing the previous contents, and are

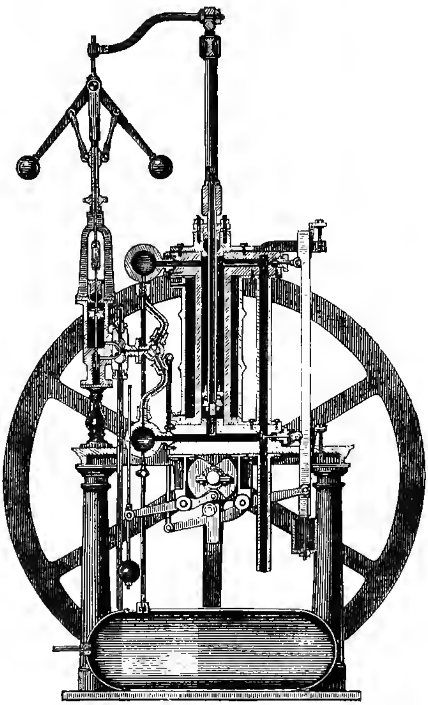


FIG. 2.—Wright's Gas-exploding Engine, 1833

ignited while the piston is crossing the dead centre. The explosion pushes the piston up or down through its whole stroke ; at the end of the stroke the exhaust valve opens and the products of combustion are discharged during the return, excepting the portion remaining in the spaces not entered by the piston. The ignition is managed by an external flame and touch-hole. The author has been unable to find whether the engine was ever made, but the knowledge of the detail essential to a working gas engine shown by the drawings indicates that it or some similar machine had been worked by the inventor. Both

cylinder and piston are water-jacketed, as would have been necessary in a double-acting gas engine to preserve the working parts from damage from the intense heat of the explosion. This is the earliest drawing in which this detail is properly shown.

William Barnett, 1838, *No.* 7615.—Barnett's inventions as described in his specification are so important that they require more complete description than has been here accorded to earlier inventors.

Barnett is the inventor of a very good form of igniting arrangement. The flame method most widely used up to about 1892 was originated by him.

Barnett is also the inventor of the compression system now so largely used in gas engines. The Frenchman, Lebon, it is true, described an engine using compression in the year 1799, but his cycle is not in any way similar to that proposed by Barnett, or used in the modern gas engine. Barnett describes three engines. The first is single-acting, the second and third are double-acting; all compress the explosive mixture before igniting it. In the first and second engines the inflammable gas and air is compressed by pumps into receivers separate from the motor cylinder, but communicating with it by a short port which is controlled by a piston valve. The piston valve also serves to open communication between the cylinder and the air when the motor piston discharges the exhaust gases.

In the third engine the explosive mixture is introduced into the motor cylinder by pumps, displacing as it enters the exhaust gases resulting from the previous explosion; the motor piston by its ascent or descent compresses the mixture. Part of the compression is accomplished by the charging pumps, but it is always completed in the motor cylinder itself.

In all three engines the ignition takes place when the crank is crossing the dead centre, so that the piston gets the impulse during the whole forward stroke.

Fig. 3 is a sectional elevation of the first engine, showing the principal working parts, but omitting all detail not required for explaining the action.

There are three cylinders containing pistons: A is the motor piston, B is the air-pump piston. The gas-pump piston cannot be seen in the section, but works in the same crosshead as B. The motor piston is suitably connected to the crank-shaft, and the other two are also connected by levers in such manner that all three move simultaneously up or down. The pump pistons, moving up, take respectively air and inflammable gas into their cylinders; upon the down stroke the gases are forced through an automatic lift valve into the receiver D, and there mix. When the down stroke is complete and the receiver is fully charged with the explosive mixture,

the pressure has risen to about 25 lbs. per square inch above atmosphere. At the same time as the pumps are compressing, the motor piston is moving down and discharging the exhaust gases from the power cylinder; it reaches the bottom of its stroke just when compression is complete. The piston valve E then opens communication between the receiver and the motor, at the same time closing to atmosphere. The motor cylinder being in free communication with the receiver, the explosion of the mixture is accomplished by the igniting cock or valve F; the pressure resulting actuates the motor

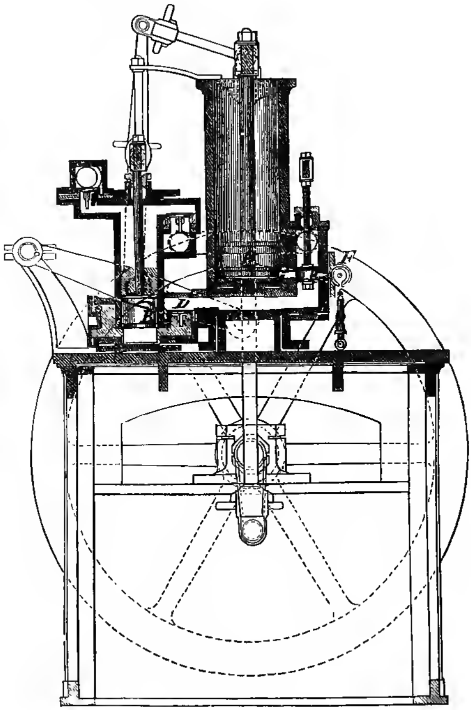


FIG. 3.—Barnett Gas Engine

piston during its whole upward stroke, the hot gases flowing through the port G precisely as steam would do. The volume of the receiver being constant, the pressure in the motor cylinder slowly falls by expansion, due to the movement of the piston, upon which work is performed, and by cooling, the pressure still existing in the cylinder when the stroke is complete depending on the ratio between the volume swept by the motor piston and the volume of the receiver.

The down stroke again expels the products of combustion, the

valve opening to atmosphere, while the compression again takes place. This cycle gives a single-acting engine. It is obvious that as the piston A does not enter the receiver it cannot displace the exhaust gases there. If means are not taken to expel these gases they must mix with the fresh explosive charge pumped in.

It is very desirable that these gases should be as completely as possible discharged. An exhausting pump is described for doing this, but in small engines it adds an additional complication; and so Barnett states that in some cases it may be omitted. The exhaust gases do not so injuriously affect the action of small gas engines.

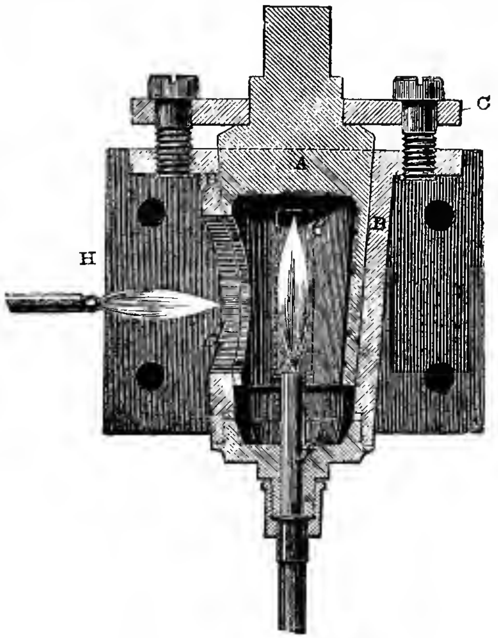


FIG. 4.—Barnett's Igniting Cock

The igniting valve is very ingenious. It is shown at Fig. 4, on a larger scale. A hollow conical plug A is accurately ground into the shell B, and is kept in position by the gland C; the shell has two long slits, *d* and *e*; the plug has one port so cut that as the plug moves it shuts to the slit *d* before opening to *e*. In the bottom of the shell there is screwed a cover carrying a gas burner *f*, which may be lit while the port in the plug is open to the air through *d*. The external constant flame H lights it. So long as the plug remains in this position the internal flame continues to burn quietly. If the plug be now turned to shut to the outer air, it opens to the slit *e*, and as that

contains explosive mixture it at once ignites. The explosion extinguishes the internal flame, but it is again lighted at the proper time when the plug is moved round. The valve acts well and is almost identical in principle with he flame-igniting arrangements of Hugon, Otto Langen, and Otto.

Barnett's second engine is identical with his first except that it is double-acting, and therefore requires a greater number of parts.

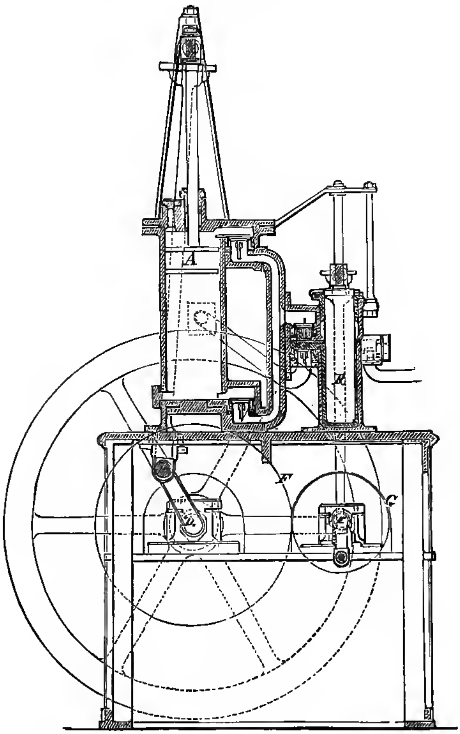


FIG. 5.—Barnett Engine

Barnett's third engine is worthy of careful description. Fig. 5 is a vertical section of the principal parts. It is double-acting. It has three cylinders, motor, air-pump and gas-pump ; the air and gas pumps are single-acting, the motor piston is double-acting. The pumps are driven from a separate shaft, which is actuated from the main crank-shaft by toothed wheels ; the wheel upon the pump-shaft is half the diameter of that on the motor shaft, so that it makes two revolutions for one of the other. The pumps therefore make one up-and-down stroke for each up or down stroke of the motor piston ;

the angles of the cranks are so set that they (pumps) discharge their contents into one or other side of the motor cylinder at every stroke; the exhaust gases are partly displaced by the fresh explosive mixture, and the motor piston completes the compression in the motor cylinder itself. When full up or down the igniting cock acts and the explosion drives the piston to the middle of its stroke; it here runs over a port in the middle of the cylinder, and the pressure at once falls to atmosphere.

A is the motor piston; B is the air-pump piston; the gas-pump piston is behind the air-pump, and is therefore not seen in the section; D is the main crank-shaft; E the pump-shaft driven from the main shaft by the wheels F and G. The engine is exceedingly interesting as the first in which the compression is accomplished in the motor cylinder, but it is not so good a machine as the first because of the difficulty of obtaining a sufficient amount of expansion.

From 1838 to 1854 inclusive eleven British patents were applied for; some were not completed, but only reached the provisional stage. Of these patents by far the most important is Barnett's; the others are interesting as showing the gradual increase of attention the subject attracted. The other names are Ador, 1838; Johnson, 1841; Robinson, 1843; Reynolds, 1844; Brown, 1846; Roger, 1853; also Bolton & Webb, making three patents for the year; for 1854 two patents, Edington and Barsanti & Matteucci. None of the proposals in these patents are really valuable or novel, being anticipated by either Street, Wright, or Samuel Brown. Robinson's is the best, being similar to Lenoir's in some of its details, and showing distinctly a better understanding of gas-engine detail.

A. V. Newton, 1855, *No.* 562.—This specification is interesting, and describes for the first time a form of igniting arrangement which came into use about 1885; it seems to be identical with the invention of the American Drake, although not described as a communication from him. It is a double-acting engine, and takes into the cylinder a charge of gas and air mixed, during a portion of the stroke, at atmospheric pressure. The igniting arrangement is a thimble-shaped piece of hard cast iron which projects into a recess formed in the side of the cylinder: it is hollow, and is kept at all times red-hot by a blowpipe flame projected into it by a small pump. When the piston uncovers the recess the explosive gases coming in contact with it ignite, and the pressure produced drives it forward.

This is the first instance of ignition by contact with red-hot metal; the proposal has often been made since then in varying forms.

Barsanti & Matteucci, 1857, *No.* 1655.—This is the first free piston engine ever proposed; instead of allowing the explosion to act directly upon the motive power shaft through a connecting rod, at

the moment of explosion the piston is perfectly free. The cylinder is very long, and is placed vertically. When the explosion occurs it expends its power in giving the piston velocity; the expansion therefore takes place with considerable rapidity, and the piston, gaining speed until the pressure upon it falls to atmosphere, moves on till the energy of motion is absorbed, doing work on the external air, lifting the piston and in friction. When the energy is all absorbed in this manner it stops; it has reached the top of its stroke. A partial vacuum has been formed in the cylinder and the weight has been raised through the stroke. It now returns under the pressure of the atmosphere and its own weight; in returning a rack attached to the piston engages the motive shaft and drives it. The cooling of the gases as the piston descends continues and helps to keep up the vacuum.

The method although indirect is economical. Three advantages are gained by it—rapid expansion, considerable expansion (an expansion of six times is common in these engines), and also some of the advantages of a condenser.

Fig. 6 shows a vertical section of their best modification. The motor piston A working in the tall vertical cylinder B is attached to the rack C, which works into the toothed wheel D. The motor shaft E revolves in the direction of the arrow, and it is provided with a ratchet; a pall upon the wheel D engages the ratchet on the down stroke of the piston only; on the up stroke it slips freely past the ratchet. The piston A is therefore quite free to move without the shaft on the up stroke, but it engages on the down stroke. The cams F and G are arranged to strike projections upon the rack, and so raise or lower the piston. It is raised when the charge is to be taken in, and lowered when it has completed its working stroke and the exhaust gases have to be discharged. When raised the valve H is in the position shown. Air first enters the cylinder through the port I, which also serves to discharge the exhaust. After the piston has uncovered the port K the valve H shuts on I, opening at the same time on K; the gas supply then enters and mixes more or less perfectly with the air previously introduced.

A small further movement of the piston now closes the valve and the explosion is caused by the passage of the electric spark in the position indicated upon the drawing. The piston shoots up freely to the top of its stroke, to give out the work stored up usefully upon its return.

As the next engine to be described marks the beginning of the practicable stage of gas-engine development, it is advisable to summarise before proceeding.

Previous to 1860 the gas engine was entirely in the experimental stage. Many attempts were made, but none of the inventors sufficiently overcame the practical difficulties to make any of their engines

commercially successful. This was mostly due to the very serious nature of the difficulties themselves, but it was also due to the too great ambition of the inventors: they wished not only to compete with the steam engine for small powers, but for large powers. They thought, in fact, more to displace the steam engine than to compete with it.

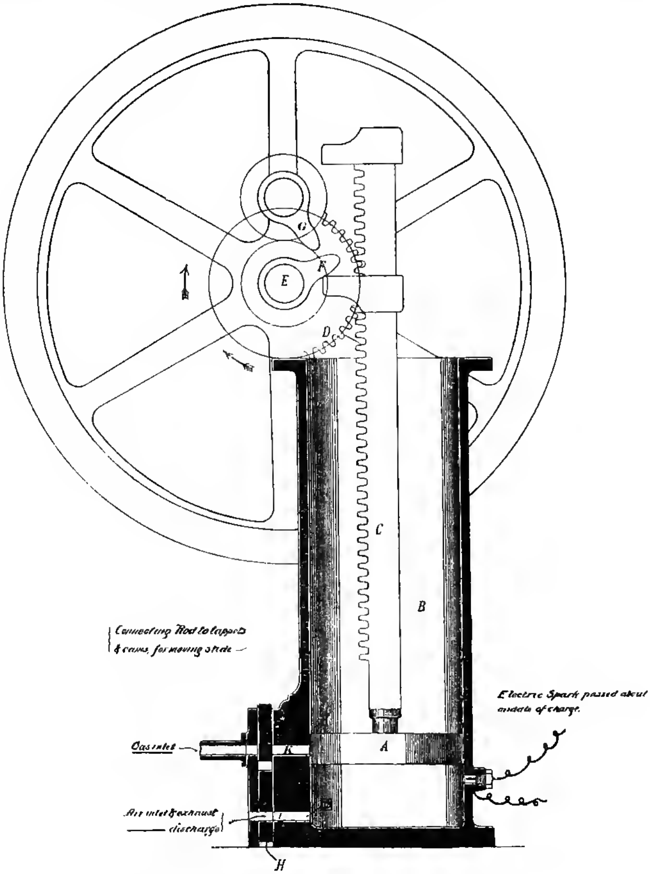


FIG. 6.—Barsanti & Matteucci Engine, 1857

This is clearly shown in many of their descriptions of the applications of their inventions.

The greatest credit is due to Wright and Barnett. Wright very closely proposed the modern non-compression system, Barnett the modern compression system. Barnett is also the originator of one of the modern flame systems for ignition. Barsanti & Matteucci follow in order of merit as the inventors of the free-piston gas engine.

Lenoir occupies the honourable position of the inventor of the first gas engine ever actually introduced regularly to public use. The engine was not strikingly novel; nothing was done in it which had not been proposed before, but its details were thoroughly and carefully worked out. It was, in fact, the first to emerge from the purely experimental stage. Lenoir's real credit consists in overcoming the practical difficulties sufficiently to make previous proposals fairly workable.

The principle is exceedingly simple and evident. The piston moves forward for a portion of its stroke, by the energy stored in the fly-wheel, and takes into the cylinder a charge of gas and air at the ordinary atmospheric pressure. The valves cut off communication, and the explosion is occasioned by the electric spark: this propels the piston to the end of the stroke. Exhausting is done precisely as in the steam engine.

The engine is simply an ordinary high-pressure steam engine with valves arranged to admit gas and air and discharge the products of combustion. Fig. 7 is an external elevation of a three-horse engine. It was first constructed in Paris in 1860 by M. Hippolyte Marinoni. In Moigno's 'Cosmos' of that year it is stated that two engines were in course of manufacture—one of six horse-power, the other of twenty.

The early statements of its economy were ludicrously inaccurate. A one-horse-power engine consumed, it was said, but three cubic metres (106 cubic feet nearly) of coal gas in twelve hours' work, and therefore cost for fuel not more than one-half of what a steam engine would have done.

The actual consumption was speedily shown to be much nearer three cubic metres per effective horse-power per hour.

Notwithstanding the high consumption, the engine had many good points: its action was exceedingly smooth; no shock whatever was heard from the explosion. Indeed it is quite impossible when watching the engine in motion to realise that regular explosions are occurring. The motion is as smooth and silent as in the best steam engine.

In the 'Practical Mechanics' Journal' of August 1865 there is an article describing the progress made by the engine since the date of its introduction, from which it appears that in France from 300 to 400 engines were then at work, the power ranging from half-horse to three-horse.

The Reading Ironworks Company, Limited, at Reading, undertook the manufacture for this country. One hundred engines were made and delivered by them; several of them have continued at work till now. Notably one engine inspected by the author at

Petworth House, Petworth, worked for twenty years pumping water, and is even yet in good condition.

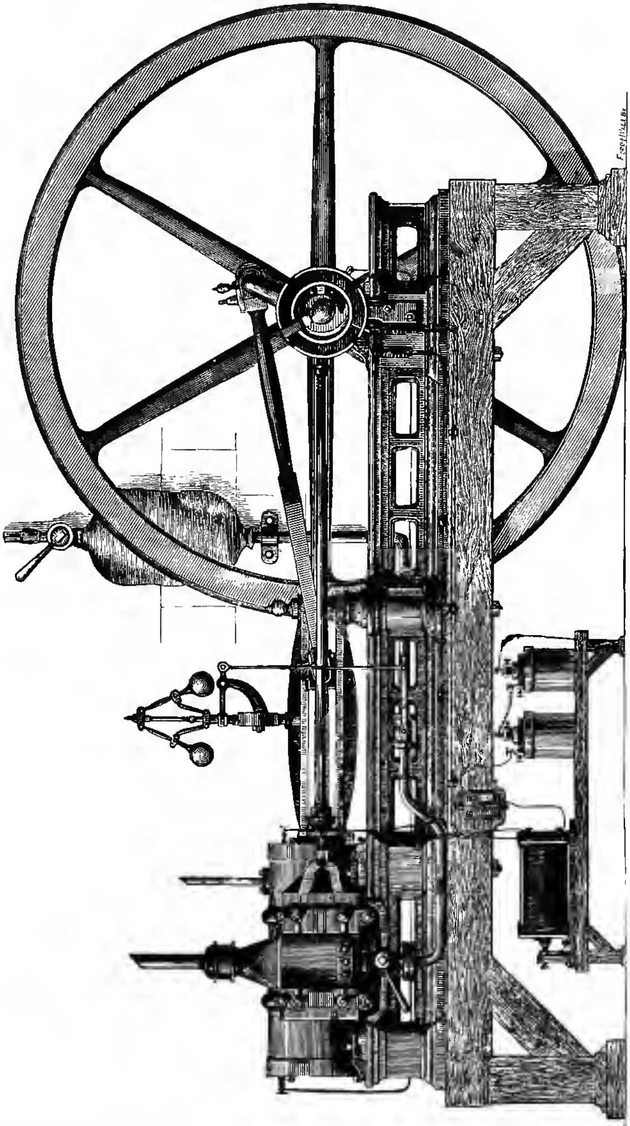


Fig. 7.—Lenoir's Gas Engine

The work performed by the engines was multifarious in its character: printing, pumping water, driving lathes, cutting chaff, sawing stone,

polishing marble—in fact, wherever from one-half to three horse-power was sufficient.

Lenoir built an experimental road carriage propelled by one of his engines which was proved to have repeatedly circled round the works where it had been constructed in the Rue de la Roquette, in Paris, at the point of turning to Vincennes. It also made a trip from Paris to Joinville-le-Pont and returned within three hours.

A two-horse-power Lenoir engine was also placed in a boat, which is said to have run between Paris and Charenton several times a week for two years. The power obtained was found to be too small, and great difficulty was experienced with cooling water on the road carriage.

The fuel used was a light volatile hydrocarbon, vaporised by a surface evaporating device. The fuel was similar to petrol, though not known by that modern name.

Lenoir's patent in this country was obtained by J. H. Johnson, 1860, No. 335. It describes very closely the engine as manufactured both in France and England. The subsequent patent 1861, No. 107, does not seem to have been carried into effect.

These specifications contain many erroneous ideas, showing the notions then prevalent among inventors of the nature of gaseous explosions. Lenoir erroneously supposed that the economy of his engine would be improved if he could obtain a slower explosion. He evidently thought that the power imparted to the piston by explosion was similar in nature to a sudden blow—a rapid rise of pressure, and a fall nearly as rapid. He therefore attempted to avoid explosion by such expedients as stratification and injection of steam or water spray. The stratification idea he very clearly expressed in his second specification, stating that 'the object of preventing the admixture of air and gas is to avoid explosion.' It is somewhat extraordinary to find notions so erroneous common at a time when Bunsen's work had clearly proved the continuous nature of the combustion in gaseous explosions, and when Hirn had made experiments which showed that the heat evolved by explosion in a gas engine was only a small part of the total heat of the combustion, the heat which did not appear during explosion being produced during expansion.

Other speculations on the cause of the uneconomical working of the engine were frequent, but the true reason was fully explained by Gustav Schmidt in a paper read before 'The Society of German Engineers' in 1861. He states: 'The results would be far more favourable if compression pumps, worked from the engine, compressed the cold air and cold gas to three atmospheres before entrance into the cylinder; by this a great expansion and transformation of heat is possible.'

This opinion became common at this time. Compression engines

were proposed with great clearness and a full understanding of the advantages to be gained.

Million, 1861, *No.* 1840.—This Frenchman had exceedingly clear ideas of the advantages of compression; he evidently considers himself as the first to propose its use in a gas engine, apparently unaware of the existence of Barnett's engine already described. He claims the exclusive right to use compression in the most emphatic language.

The first engine described is exactly what Schmidt asks for. Separate pumps compress the air and gas into a reservoir, from which the movement of the motor piston, during a portion of the stroke, withdraws its charge under compression. Ignition is accomplished by the electric spark, and the piston moves forward under the high pressure produced. He states:

'In ordinary air engines the operation of the motive cylinders is analogous to that of the pumps, the result being that there are two cylinders, which act in directions contrary to each other, and that the pump, which is an organ of resistance, even works at a greater pressure than that of the motive cylinder, which is an organ of power. Thus these engines are very large in proportion to their power. On the contrary, by employing gases under the conditions above explained, these engines will exert great power in proportion to their dimensions. The sudden ignition of the gases in the motive cylinder causes the latter to work at an operative pressure much greater than that of the pumps.'

The advantage of compression in a gas engine could not be more fully and clearly stated. But he goes even a step further; he sees that the portion of the motor piston stroke spent in taking in the charge under compression is a disadvantage, and he proposes to make the whole stroke available for power by providing a space at the end of the cylinder in which the gases are compressed.

'Instead of introducing the cold gases into the cylinders, during a portion of the stroke and igniting them afterwards, when the induction ceases . . . another arrangement might be adopted. The motive cylinder might be made longer than necessary in order that the piston should always leave between it and the end of the cylinder a greater or less space, according to the pleasure of the constructor, such as one-fourth or one-third, more or less, of the volume generated by the motive piston. This space is called by the inventor a cartridge. On opening the slide valve the gases could be allowed to enter suddenly from the pressure reservoir into this cartridge towards the dead point, and this induction having ceased an electric spark would ignite the gases in the cartridge by which the driving piston would be set in motion.'

Such an engine would resemble in its action the best modern compression engines. The difficulties of ignition, however, are too considerable to be overcome without further detail.

The compression idea at this date was evidently widely spread, because it again crops up in a remarkably clever pamphlet by M. Alph. Beau de Rochas, published in Paris in 1862. He advances a step further than Million, and investigates the conditions of greatest economy in gas engines using compression, with reference to volume of hot gases and surfaces exposed. He states that to obtain economy with an explosion engine four conditions are requisite :

1. The greatest possible cylinder volume with the least possible cooling surface;

2. The greatest possible rapidity of expansion;

3. The greatest possible expansion ; and

4. The greatest possible pressure at the commencement of the expansion.

In using boiler tubes, he states, the efficiency of the heat transmitted increases with reduction in the diameter of the tubes. In the case of engine cylinders, therefore, the loss of heat of explosion would be in inverse ratio to the diameter of the cylinders.

Therefore, he reasons, an arrangement which, for a given consumption of gas gives cylinders of the greatest diameters, will give the best economy, or least loss of heat to the cylinder. One cylinder only must be employed in such an engine.

But loss of heat depends also upon time ; cooling, therefore, will be proportionately greater as the working speed is slower.

The sole arrangement capable of combining these conditions, he states, consists in using the largest possible cylinder, and reducing the resistance of the gases to a minimum. This leads, he states, to the following series of operations.

1. Suction during an entire outstroke of the piston.

2. Compression during the following instroke.

3. Ignition at the dead point and expansion during the third stroke.

4. Forcing out of the burned gases from the cylinder on the fourth and last return stroke.

The ignition he proposes to accomplish by the increase of temperature due to compression. This he expects to do by compressing to one-fourth of the original volume.

In our own country the late Sir C. W. Siemens proposed compression in 1862. The idea was exceedingly widely spread, as is evident from those numerous and independent inventions. The practical experience to enable it to be successfully effected had yet to be created, however, and this took many years of patient work.

The igniting arrangement was the first weak point requiring improvement. The electrical method of Lenoir was exceedingly delicate and troublesome.

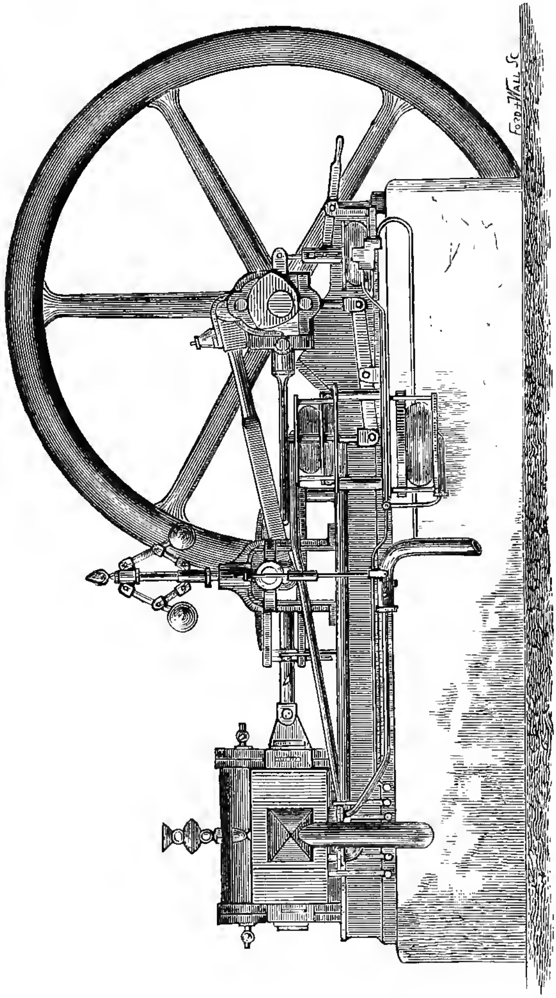


Fig. 8.—Hugon Engine

Hugon's engine, produced in 1865, was similar to Lenoir's; but the igniting was accomplished by flame, a modification of Barnett's, 1838, using a slide valve instead of a lighting cock. The flame ignition was certain and easily kept in order. In other points the engine was a great improvement upon its predecessor. The lubrication

was improved by injecting water into the cylinder and the cooling water jacket was better arranged. As a result the consumption of gas was reduced.

Fig. 8 is an external elevation of the Hugon engine.

Mr. Otto now appears upon the scene. Before him much had been done in inventing and studying engines, but it remained for him by sheer perseverance and determination of character, to overcome all difficulties and reduce to successful practice the theories of his predecessors.

In 1867 Messrs. Otto & Langen exhibited at the Paris exhibition of that year, their free piston engine, exterior elevation shown at fig. 19. It was absolutely identical in principle with the previous invention of Barsanti and Matteucci, but the details were completely and successfully carried out. The Germans succeeded commercially and scientifically where the Italians completely failed.

Flame ignition was used and great economy was obtained, a half-horse engine, according to Professor Tresca, giving over half-horse power effective, on a gas consumption at the rate of 44 cubic feet per effective horse-power per hour. This is less than half the consumption of Lenoir or Hugon; accordingly the prejudice excited by the strange appearance and noisy action of the engine did not prevent its sale in large numbers. It completely crushed Lenoir and Hugon, and held almost sole command of the market for ten years, several thousands being constructed in that period.

The Brayton gas engine appeared in America in 1873 but although more mechanical than any free piston engine, its economy was insufficient to enable it to compete. It was better than Lenoir or Hugon, but not nearly so good as Otto & Langen.

In this engine there are two cylinders, compressing pump and motor. The charge of gas and air is drawn into the pump on the outstroke and compressed on the return into a receiver; the pressure usual in the receiver varies from 60 to 80 lbs. per square inch above atmosphere. The motor cylinder takes its supply from the receiver, but the mixture is ignited as it enters, a grating arrangement preventing the flame from passing back; the mixture, in fact, does not enter the motor cylinder at all; what enters it, is a continuous flame. At a certain point the supply of flame is cut off and the piston, moving on to the end of its stroke, expands the volume of hot gases to nearly atmospheric pressure before discharge.

Fig. 9 is an external view of the engine. Figs. 10 and 11 are sections of the motor and pump cylinders. The action is as follows: The engine is single-acting, receiving one impulse for every revolution; like all gas engines it depends upon the energy stored up in the flywheel to carry it through those parts of its cycle where the work is

negative. The two cylinders are inverted, and are attached to a beam rocking beneath them by connecting rods. The beam is prolonged and connected to the crank above it by a rod; both cylinders are single-acting and the pistons are of the trunk kind. Both pump and motor cylinders are of the same diameter, but the pump is only half the stroke of the motor. The valves are actuated from a shaft running at the same rate as the main shaft and driven from it by bevel wheels. There are four valves, all of the conical seated kind—two upon the

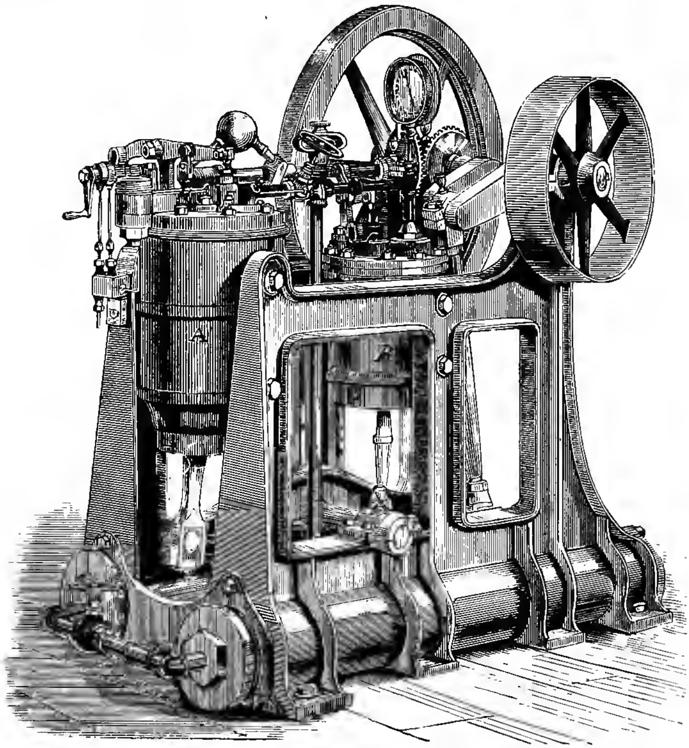


FIG. 9.—Brayton Petroleum Engine

motor, admission and discharge; two upon the pump cylinder, admission and discharge. The admission and discharge valves upon the motor are actuated from the auxiliary shaft by levers and cams, so is the pump inlet. The pump discharge valve is automatic, rising at the proper time by the pressure of compression. During the down-stroke the pump takes in the charge of gas and air, forcing it on the up-stroke into the receiver. From the receiver it is led to the power cylinder, passing by the inlet valve through a pair of

perforated brass plates with wire gauze placed between them. Through this diaphragm a small stream of mixture is constantly passing into the motor cylinder; before the engine is started, a plug is withdrawn and the current lighted; a constant flame is therefore burning under the diaphragm. The mixture enters the cylinder through this flame, lighting as it enters; at all times during the exhaust part of the stroke, as well as the admission, the stream of entering mixture from the

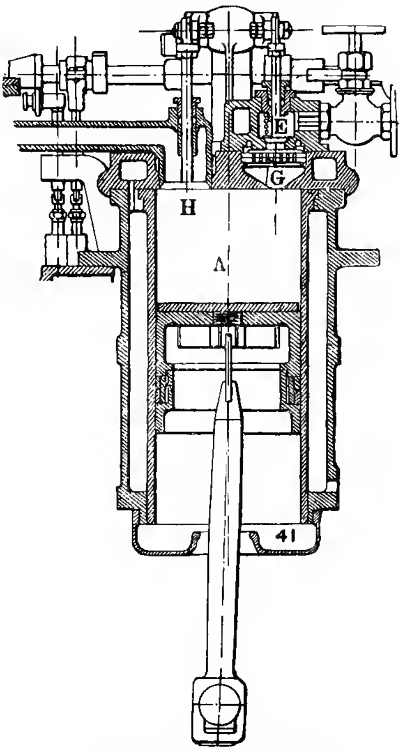


FIG. 10.—Brayton Engine Section of Motor Cylinder

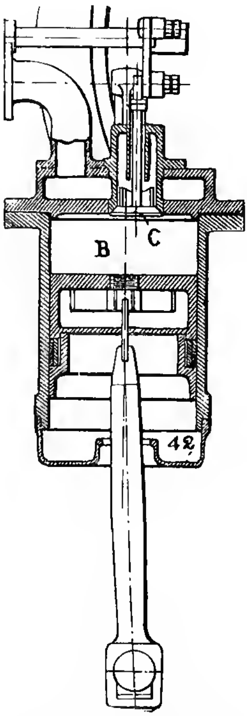


FIG. 11.—Brayton Engine Section of Pump Cylinder

receiver keeps up a small constant flame which is augmented at the beginning of the stroke, so as to fill the cylinder entirely, when the admission valve is opened. When the admission valve is closed, the by-pass keeps the flame fed with sufficient mixture to keep it alight. The pressure in the cylinder thus never exceeds that in the reservoir, and the mixture burns quietly without spreading back.

Figs. 9, 10, 11, and 13.—A is the motor cylinder; B the pump; the beam and connections require no lettering; C is the pump inlet

valve (the pump discharge, which is an ordinary lift valve, is not seen in fig. 11, but is lettered D in fig. 9); E the motor inlet; F the igniting plug, which is withdrawn when the flame is to be lit before starting the engine (see fig. 13); G is the grating in section (see fig 10); H the exhaust valve; the levers and cams are sufficiently indicated on the drawing; the small pipe and stop-cock (fig. 9) communicates at all times with the reservoir and supplies the constant flame with mixture. The engine worked well and smoothly; the action of the flame in the cylinder could not be distinguished from that of steam, it was as much within control and produced diagrams quite similar to steam.

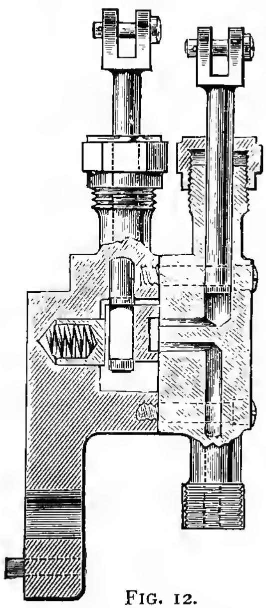
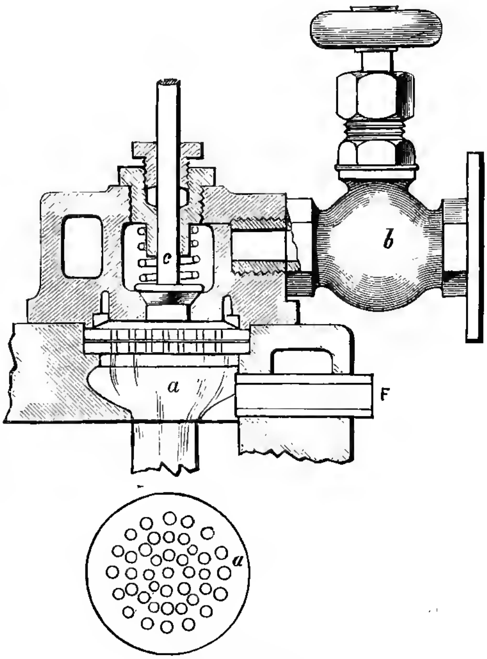


FIG. 12.
Brayton Petroleum Pump



Plan of Grating
FIG. 13.—Brayton Grating and Valve

The flame grating was the weak point; it stood exceedingly well for a time, but if by any accident the gauze was pierced in cleaning, the flame went back into the reservoir and exploded all the mixture—the engine, of course, pulled up as the constant flame having no supply was extinguished. This accident became so troublesome that Mr. Brayton discontinued the use of gas and converted his engine into a petroleum engine. The light petroleum was pumped upon the grating into a groove filled with felt, the compressing pump then

charged the reservoir with air alone. The air in passing through the grating carried with it the petroleum, part in vapour, part in spray; the constant flame was fed by a small stream of air. The arrangements were, in fact, precisely similar to the gas engine, except in the addition of the small pump and the slight alteration in the valve arrangements. The difficulty of explosion into the reservoir was thus overcome, but a new difficulty arose—the cylinder accumulates soot with great rapidity and the piston requires far too frequent removal for cleaning. The petroleum pump is an exceedingly clever little contrivance; fig. 12 shows its details. The amount of petroleum to be injected at each stroke is so small that an ordinary force-pump with clack valves would be uncertain. Brayton gets over this difficulty by substituting a slide valve driven from the eccentric.

The plunger of the pump is no larger than a black-lead pencil, yet it discharges any quantity, from a single drop per stroke up to a full throw, with unerring certainty. The plunger also is driven from an eccentric. Both eccentrics are in one piece and rotate on the end of the auxiliary shaft, driven by a pawl when the engine is in motion; to allow of starting, the pump can be moved by a hand-crank independently. To start, the air reservoir is filled, if not already full, by turning the engine round by hand; the plug F is then withdrawn and a little petroleum thrown upon the diaphragm by a few turns of the pump. The cock on the small pipe is then opened and a stream of air flowing from the reservoir vaporises the petroleum; it is lit at G, and the flame having enough air for combustion retreats to the grating and remains burning within the cylinder. The plug is then inserted, the starting cock opened, and the engine starts. The flame remains alight during the whole time the petroleum continues to be supplied.

The valves act well, and the motor cylinder does not suffer from the action of the flame so long as it is kept reasonably clean. If the soot, however, is allowed to accumulate, it speedily cuts up.

The late Prof. Thurston, of the Stevens Institute of Technology, tested a Brayton gas engine in New York in the year 1873.

The following extracts are from his report:

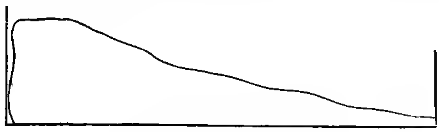
'The operation of the engine is precisely similar, in the action of the engine proper and in the distribution of pressure in its cylinder, to that of the steam engine. The action of the impelling fluid is not explosive as it is in every other form of gas engine of which I have knowledge.

'Upon the opening of the induction valve, the mixed gases enter, steadily burning as they flow into the cylinder, and the pressure from the commencement of the stroke to the point of cut-off, as is shown by the indicator diagrams, is as uniform as that observed in any steam engine cylinder. The maximum pressure exerted during my experi-

mental trial, and while the engine was driving somewhat more than its full rated power, was about 75 lbs. per square inch at the beginning of the stroke, gradually diminishing to 66 lbs. per square inch at the point of cut-off, where the speed of the piston was nearly at a maximum, and then declining in accordance with the law governing the expansion of gases.

'Complete combustion is insured by thorough mixture. This is accomplished by taking the illuminating gas and air, in proper proportion, into the compressing pump together, and the mixture here made becomes more intimate in the reservoir, and in its progress towards the point at which it does its work. The constantly burning jet already described insures prompt ignition on entering the cylinder.

'. . . the engine rated at 5 HP developed, as a maximum, rather more than its rated power. Its mean power during the test, as determined by the dynamometer, was 3·986 HP, the indicator showing at



Max. press. 68 lbs. per sq. in.

FIG. 14

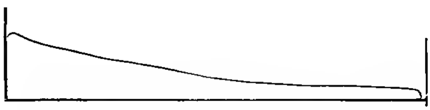


FIG. 15.—Diagrams from Brayton's Gas Engine

that time 8·62 HP developed in the cylinder. The amount of gas consumed averaged 32·06 cubic feet per IHP per hour.

'The excess of indicated over dynamometric HP is to be attributed to the work of driving the compressing pump and to the friction of the machine.

'The greater portion of this appears both on debit and credit side of the account, since, although expended in the compressing pump, it is restored again in the driving cylinder.'

The consumption of 32·06 cubic feet per IHP per hour is incorrect; it is obviously unfair to include the pump diagram in the gross power. The author has tested an engine of similar construction and dimensions; he finds the friction of the mechanism to be about 1-horse; adding this number to the dynamometric power of Prof. Thurston, the legitimate indicated power may be taken as 5 HP, the consumption is therefore $\frac{8{\cdot}62 \times 32{\cdot}06}{5{\cdot}0} = 55{\cdot}2$; and the gas per brake HP per hour

is $\frac{8{\cdot}62 \times 32{\cdot}06}{3{\cdot}986} = 69{\cdot}3$. These numbers, although showing improvement upon the Lenoir and Hugon, prove that the engine was much inferior in economy to the Otto & Langen engines.

Mr. H. McMutrie, Consulting Engineer at Boston, took diagrams from an engine of similar dimensions which confirm these results.

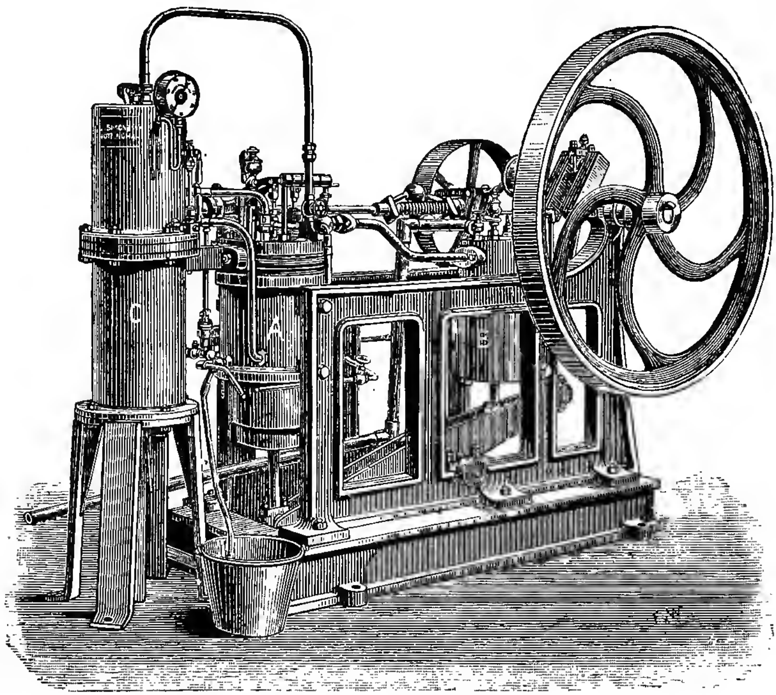


FIG. 16.—Simon Engine

Fig. 14 is the diagram taken with full load, fig. 15 the diagram from the motor with no load on, the power being just sufficient to overcome friction and pump losses.

FULL LOAD DIAGRAM

Area of piston	50·26 sq. ins.
Speed of piston	180 ft. per min.
Mean pressure	33 lbs. per sq. in.
Pressure in reservoir	75·4 lbs. per sq. in.
Initial pressure in cylinder	68 lbs. per sq. in.
Gross power developed	9 HP.

No Load Diagram

Speed of piston	180 ft. per min.
Mean pressure	18 lbs. per sq. in.
Friction and other resistance	4·87 HP.
Net available power	9 − 4·87 = 4·13

This power agrees closely with the actual determination by dynamometer.

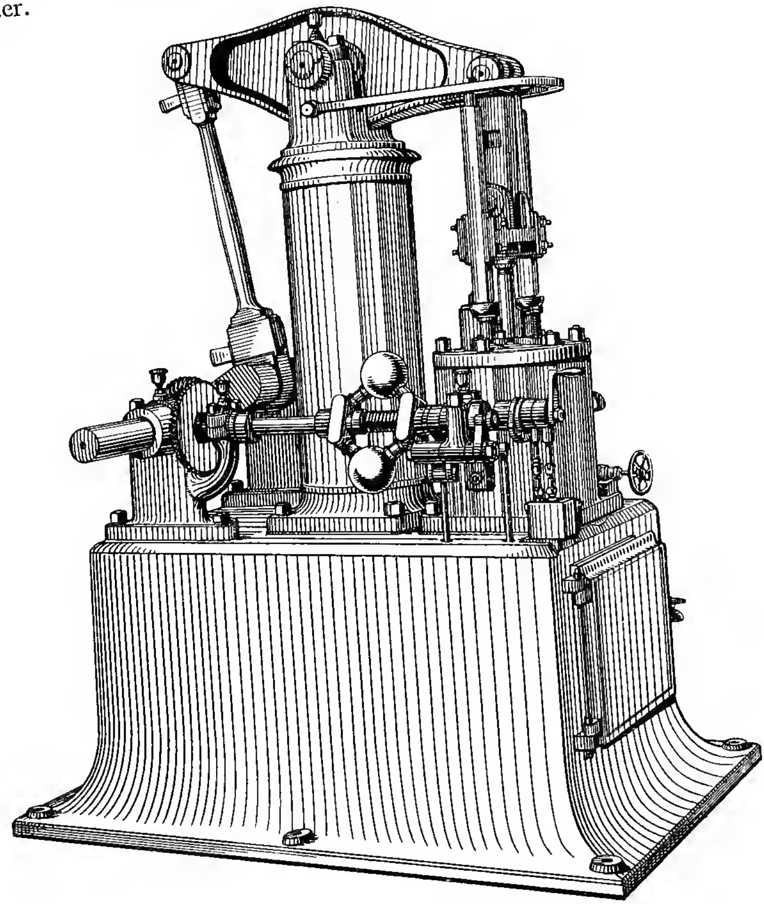


Fig. 17.—Brayton Beam Overhead Engine

Messrs. Simon, of Nottingham, introduced the Brayton engine to England in a slightly altered form as a gas engine. In addition to the ordinary arrangements of the engine they attempted to gain

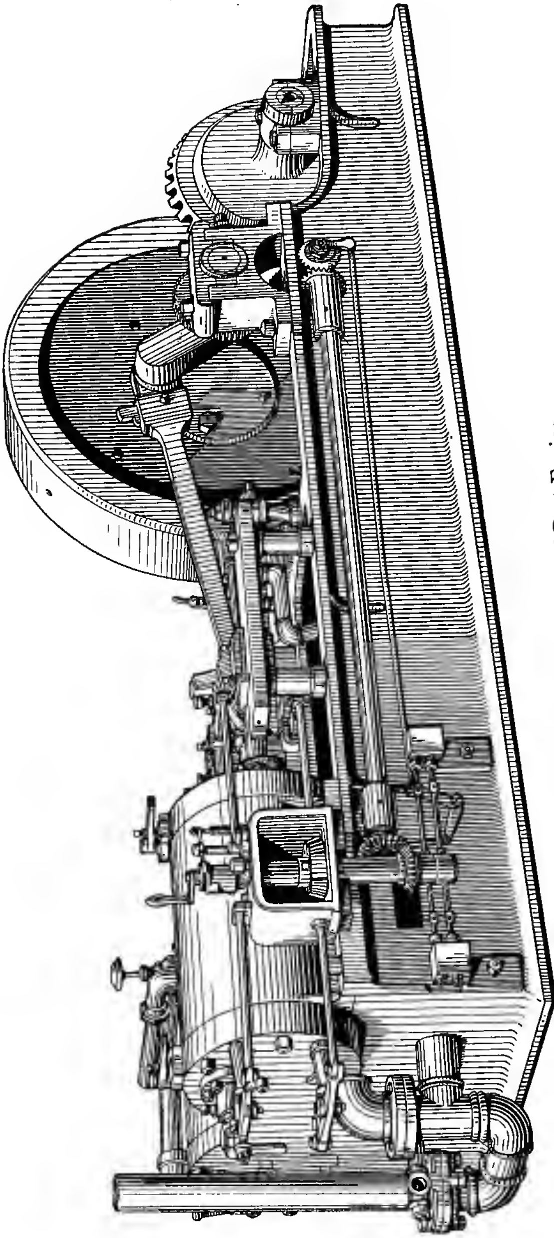


FIG. 18.—Brayton Horizontal Boat Engine

increased economy, by causing the waste heat passing into the water jacket, and the heat of the exhaust gases, to be utilised in raising steam.

Fig. 16 is an external view of the engine as exhibited at the Paris Exhibition of 1878. A is the motor, B the pump, and C the added boiler; the steam was raised in it and the water jacket. The engine, although instructive, did not successfully overcome the difficulties which caused the abandonment of the Brayton as a gas engine.

Brayton constructed this engine in America also in the beam overhead style and horizontal and inverted vertical. He applied the inverted vertical engine to a tram-car, but did not succeed in running it commercially.

The horizontal engine he applied to a boat. Two of the boats were in use upon the Hudson for some years.

Fig. 17 is an external view of a beam overhead engine, and fig. 18 is a similar view of a horizontal engine as applied to one of the boats.

Brayton was enthusiastic and indefatigable, and spent most of his life in his many experiments; ultimately he abandoned his American attempts and crossed over to England, and died in Leeds while engaged with experiments on a new oil engine at a large works there. His perseverance deserved a better reward. No one, however, has yet succeeded in carrying his type of engine further than he did.

In 1876 Dr. Otto superseded his former invention by the production of the 'Otto Silent' engine, now known all over the globe. It is a compression engine, using the precise cycle described in 1862 by Beau de Rochas, but carried out in a most perfect manner and using a good form of flame ignition—a modified Otto & Langen valve in fact. The economy is greater than that of any previous engine, one indicated horse being obtained upon 20 cubic feet of gas, or one effective horse upon 24 to 30 cubic feet per hour.

This engine has established gas engines upon a firm commercial basis.

Strangely enough, although Dr. Otto was the greatest and most successful gas engine inventor who has yet appeared, he adhered to Lenoir's erroneous ideas, and in his specification 2081 of 1876 he attributed the economy of his machine to a slow explosion caused by arrangement of gases within the cylinder.

The compression, which is the real cause of the economy and efficiency of the machine, he seemed to consider as an accidental and unessential feature of his invention.

Dr. Otto deserved his success; he had fought hard and long for it. He began his work in 1854, attained his first success in 1866, and his epoch-making advance in 1876.

He was born at Holzhausen, in Nassau, in 1832, and died in 1891.

He devoted his whole life to the study and development of the gas engine, beginning at the age of twenty-two years and continuing steadily on until his death—thirty-seven years of persistent work. During that long working period he experimented with nearly every form and cycle of engine. His early work dealt mainly with the atmospheric type in varied forms, but after 1876 he devoted himself to the four-cycle compression engine, and he patented many modifications, including a compound gas engine, and another engine in which the whole of the exhaust products were expelled by an auxiliary piston.

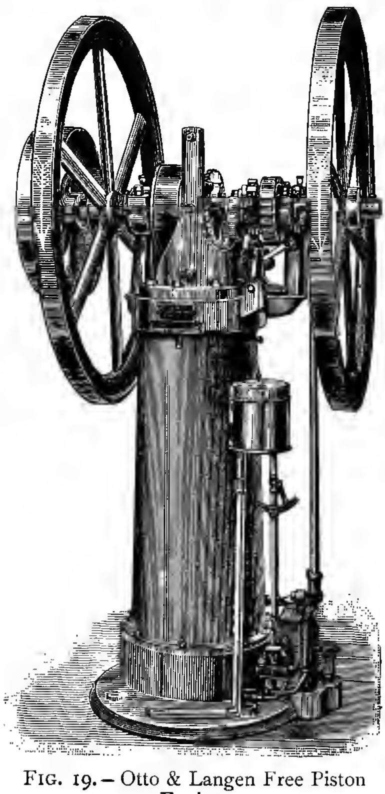


FIG. 19.—Otto & Langen Free Piston Engine

His death at the comparatively early age of fifty-nine was a great loss to the gas engine industry, but he had accomplished the heavy pioneering work for the world, and other hands took up the development of the four-cycle engine in many directions.

Daimler originated small high-speed engines consuming the light hydrocarbons which are now known as petrol; his first small engine was produced in 1883. It used a surface carburetter and dispensed with the slide ignition, substituting an open tube. These Daimler engines ultimately developed into the modern petrol engine which performs such an important part in the life of all nations at the present time. These little engines, although internal combustion engines of the old type, have developed so many interesting problems of carburettor, ignition, piston speed, and so forth, that a new science has sprung up in the course of their development which has produced motors of marvellous lightness for a given power, and made not only mechanical road locomotion a startlingly rapid success, but has brought mechanical flight just within the bounds of practicability.

Daimler was born at Schoendorf, in Würtemberg, in 1834, and he died at Canstatt in 1900, at sixty-six years of age.

He served his time with leading German engineering firms, then came to England and entered the works of Sir J. Whitworth. He joined Dr. Otto at Deutz on the Rhine in 1870, and was with him when the 'Gas Motoren Fabrik' of Cologne was formed, at which great factory he was managing director from 1872 to 1882, when he retired from the Otto works for the purpose of devoting himself to the small high-speed light oil engine.

In 1886 he tried his first motor bicycle, and on March 4, 1887, he ran for the first time a motor propelled car. About 1887, however, he had succeeded in propelling launches and other vessels on continental canals, and his engines were used for that purpose to a considerable extent. In 1889, Messrs. Panhard & Levassor made arrangements with him to manufacture motor carriages in France.

The Daimler Canstatt works became famous, and Daimler's work had undoubtedly a great share in the progress towards the modern use of petrol engines both on land and water.

Another line of development was found in the adaptation of the Otto cycle to the use of the heavier oils—that is, oils such as are used in lamps with a high flashing point. Priestman of Hull, in England, made the first success with an engine which had a spray vaporiser and electric ignition. Priestman's work began in about 1885, and resulted in the introduction of many heavy oil engines applied to all purposes on land and water. Messrs. Priestman first exhibited a 4 HP petroleum engine at the Nottingham meeting of the Royal Agricultural Society in use with ordinary lamp oil. They exhibited a 6 HP portable oil engine at the Windsor Meeting in 1888, and obtained a prize at the Plymouth meeting in 1890.

Later, Messrs. Hornsby introduced an engine known as the Hornsby-Ackroyd. This engine was the invention of Mr. Stuart Ackroyd, and it applied successfully for the first time the idea of igniting and vaporising the oil by means of the hot walls of the combustion chamber. This Hornsby-Ackroyd engine has proved to be the most successful of its type, and it is used in very large numbers both for land and marine work.

Most leading makers of gas engines in England, America, and on the Continent now manufacture both heavy and light oil engines as part of their ordinary business. Messrs. Hornsby exhibited the engine at the Royal Agricultural Society Cambridge meeting in 1894, and obtained a prize.

A most important development of the heavy oil engine is found in the Diesel engine, in which air alone is compressed in the engine cylinder to such a pressure as to heat it above the ignition point of heavy oil. When the air is at this high pressure and temperature the heavy oil is injected into it by air compressed to a still higher

pressure. The oil spray ignites as it enters and so power is produced. Diesel began his work about 1892, and by determined perseverance he has produced a most interesting engine which is now used in considerable numbers and of high powers. Diesel's first engine was produced about the year 1895.

In 1878 Mr. J. Emerson Dowson designed a complete pressure gas plant in which he intended to make a fuel gas strong enough and clean enough for factory and domestic work, and in 1881 he applied his apparatus to the supply of a gas engine. This he did at the York meeting of the British Association in 1881. The plant consisted of a closed producer containing anthracite, a boiler to raise steam, a cooler and scrubber, and a gas holder. The fuel in the producer was first raised to incandescence by means of a blower. The steam from the boiler passed through a jet blower, by which a mixture of air and steam was passed through a grate upwards through the ignited anthracite. Carbonic oxide and hydrogen were formed and passed with the nitrogen of the air through the cooler and scrubber to the gas-holder, from which the engine was fed. The engine was of 3 HP, the first engine ever operated with producer gas. At that time (1881) the largest gas engine which had been built was about 20 brake horse-power, it was called by the engineering journals of the time 'a king of gas engines.' Mr. Dowson's invention had a profound effect on the development of the gas engine. It speedily led to the construction of larger and larger engines because his method supplied fuel gas at a much lower cost per heat unit than the towns' coal gas of the time.

Dowson's pressure producers were gradually increased in dimensions as years passed on till now any desired power may be obtained. From the pressure producer sprang the suction producer, first placed on the market in practical form in 1894 by M. Benier of Paris, but then presenting many difficulties which were not removed till about nine years later, when Mr. Dowson and others placed effective suction plants in use in considerable numbers. Among the most successful of these more recent suction plants are those of the National Gas Engine Company, Ltd., and Messrs. Crossley Bros., Ltd. Nearly all makers of gas engines now manufacture suction plants. Broadly, suction plant differs from pressure plant in dispensing with the steam boiler and the gas holder, and causing the engine itself to draw through the producer the air and steam required for its operations. Suction plants have further cheapened the gas supply for gas engines to such an extent that the fuel cost is now reduced in many cases to one-tenth of a penny per brake horse-power per hour. Suction gas plants are now made in units of several hundreds of horse-power.

Pressure gas plants are now made of very large dimensions in accordance with Dr. Mond's inventions, and in smaller sizes by Messrs.

Crossley, which use bituminous fuel; but so far no bituminous fuel suction plant has yet appeared, although many workers are busy on the problem. Suction producers are so far practically confined to anthracite and coke. The gas producer is advancing rapidly in public favour, and undoubtedly it has greatly enlarged the field of the gas engine. Town's coal gas, however, remains the principal and the best fuel for the gas engine, when it can be obtained at a reasonable price; and the gas engineers of Britain are now fully alive to the importance of gas power, as about one-fifth of the whole town's gas-supply is used in engines. They have made great strides in recent years in the supply of cheap and good coal gas, so that several towns now supply coal gas at 1*s.* per thousand cubic feet for engines, and many do so at 1*s.* 6*d.*

In 1895 the late Mr. B. H. Thwaite demonstrated that the so-called waste gases from blast furnaces could be used in gas engines, and the demonstration has undoubtedly led to the design and construction of the very large gas engines now becoming common on the Continent, in America, and in this country. It appears from his experiments that the surplus gas from the blast furnaces of Britain is capable of supplying at least three-quarters of a million horse-power continuously day and night, and it is calculated that in America nearly three million horse power is available from this source. Thwaite's system was put into operation in 1895 at the Glasgow Iron Works, and it was also successfully applied near Barrow-in-Furness. For many reasons the system did not take immediate root in England, but in 1898 the Société Cockerill, of Seraing, near Liège, applied an engine designed by M. Delamere-Deboutteville to utilise blast-furnace gas of 110 British thermal units per cubic foot heat value. The engine had a single cylinder of 31·5 ins. diameter and 3 feet 3½ ins. stroke; it indicated 213 HP at 105 revolutions per minute. Before dealing shortly with the development of these large gas engines it is desirable to go back again to 1876 to give a brief account of the principal mechanical changes made in the four-cycle gas engine from that time to 1908, to consider the leading points in the progress of the two-cycle engines, and to discuss the effect of the true theory of the gas engine on its practical development.

Comparing the 1877 Crossley Otto engine with one built by the same firm in 1892, it will be found that the two to one bevel gear driving the valve shaft has been replaced by the skew gear, that the slide valve ignition has been replaced by an incandescent tube igniter, and that all engine valves are of the conical seated lift type. In the later engine of 1908 the main mechanical difference is in the use of the low-tension electrical igniter instead of the incandescent tube. These changes have been accompanied by many obvious improvements

in design, by better shaped combustion spaces and much higher compression pressures. The changes do not appear great, but they led to a notable improvement in thermal efficiency and in power obtained for given cylinder dimensions. The thermal efficiency is more than doubled; in 1876 it was possible to obtain 16 per cent., and in 1908 35 per cent. has been certainly attained.

These changes were accompanied by a gradual increase of power. In 1878, 3 HP was considered quite a large gas engine, it was the largest ever built of the Otto and Langen type; in 1881 a 20 horse engine was a 'king of gas engines'; and in 1898 the largest engine built indicated 220 HP. Writing in that year the author stated:

'It is evident that the large gas engines of ten years hence will differ as much from the large gas engine of to-day as does the latter from the engine of 1886 or 1887. There can be little doubt that in ten years, gas engines of 1000 horse-power will be as common as engines of 100 horse-power are now.'

While the Otto cycle engine was developing, inventors were hard at work on the two-cycle engine. In Britain this work fell mostly upon Clerk, Robson, and Atkinson, while on the Continent the most persevering and determined worker was Koerting.

Clerk began work on the gas engine at the end of 1876. His first patent was taken out at the beginning of 1877, and dealt with an engine of the air-pressure vacuum type in which the explosion compressed air into a reservoir and caused a partial vacuum in the explosion chamber and a vessel connected with it. Clerk's next patent was taken out in 1878 (No. 3045 of 1878), and the engine then described was exhibited at the Royal Agricultural Show at Kilburn, London, in 1879. In this engine a pump compressed a mixture of gas and air into a reservoir at the full pressure required for compression; the mixture under compression was admitted to the motor cylinder during the first part of its stroke, cut off, and then ignited by a platinum igniter, the piston driven forward by the explosion and expansion and exhausting performed on the return stroke.

The engine gave 3 brake horse-power, and it was the first compression explosion engine which was ever run giving one impulse for every revolution of the engine. This engine did not reach the market.

The engine best known as the Clerk cycle engine was patented in 1881 (No. 1089 of 1881), and exhibited at the Paris Electrical Exhibition of that year. In it the pump was used as a displacer only, and the mixture was transferred to the motor cylinder at some 4 lbs. per square inch above atmosphere, and the entering charge displaced the exhaust gases by way of ports overrun by the piston. This was the first engine giving an impulse for every revolution where the exhaust discharge was timed and controlled by the motor piston

only. This type of engine came largely into use as built by Sterne & Co., the Campbell Gas Engine Company, and many other makers, a modification of it was also largely made and sold by the Stockport Company. It fell out of use about 1890, when the Otto patent of 1876 lapsed and nearly all engineers adopted the Otto cycle. It is much in use, however, now as the Koerting engine for large gas engines.

Other engines of Clerk are described later in this work.

Robson began work on gas engines in 1877. His first patent is dated No. 2334 of 1877, and he produced an engine under patents 1879 and 1880 in which the front of the cylinder is enclosed and used as a pump to force gas and air into a reservoir at about 6 lbs. per square inch above atmosphere; the piston overran ports in the cylinder, but the exhaust was not timed by it; a separate exhaust valve was used which opened and closed at the proper time. This engine was built by Messrs. Tangye, and exhibited by them at the end of 1880.

Atkinson began work on the gas engine in 1878; his first patent is dated 1879, No. 3213. This engine was of the type exhibited by Clerk at Kilburn. Mr. Atkinson was indefatigable in the production of two-cycle engines, and he built a most ingenious engine called by him the 'Differential engine' which he exhibited at the Inventions Exhibition, London, in 1885. Later he produced another engine, which he called the 'Cycle engine'; this engine was proved to be the most economical of all those tested at the Society of Arts trials of motors for electric lighting in 1888-89. A fuller account of these two-cycle engines is given in a special chapter, so they need not be further dealt with here.

Broadly, however, the position at the present date is this: that Otto cycle—that is, four-cycle—engines with few exceptions monopolise the field of the smaller gas engines; while for large engines, the two-cycle or Clerk cycle engines compete on much more equal terms.

The general principles developed in this work explaining the causes of the economy of the modern gas engine were first enunciated by the author in a paper read before the Institution of Civil Engineers in April 1882.[1]

He then classified gas engines in three great groups:

Type 1.—Explosion, acting on piston connected to crank. (No compression.)

Type 2.—Compression, with increase of volume after ignition, but at constant pressure.

Type 3.—Compression, with increase in pressure after ignition, but at constant volume.

[1] 'The Theory of the Gas Engine,' by Dugald Clerk: *Minutes, Institution Civil Engineers, London.* Paper No. 1855. April 1882.

It was proved that under comparable conditions the relative theoretic efficiencies of the three types were

Type 1 = 0·21
Type 2 = 0·36
Type 3 = 0·45

It was also shown that in the actual engines the real efficiency could not be so high as the theoretic, mainly because of the large proportion of heat lost through the sides of the cylinder, by the exposure of the flame which filled the cylinder to the comparatively cold enclosing walls. A balance-sheet was given showing the disposal of 100 heat units by a compression engine. Of the 100 heat units, 17·83 were converted into indicated work, 29·28 were discharged with the exhaust gases, and 52·89 units passed through the sides of the cylinder into the water jacket.

The economy of the Otto engine over its predecessors, the Lenoir and Hugon engines, was clearly proved to be due to the fact of its using compression previous to explosion.

These conclusions were very generally accepted by scientific and practical men who had studied the subject, and in February 1884 the late Prof. Fleeming Jenkin, then Professor of Engineering at the University of Edinburgh, delivered a lecture at the Institution of Civil Engineers in London, on 'Gas and Caloric Engines.'[1] He had recalculated the efficiencies due to compression, with the result of corroborating the present writer's conclusions. He states:

'If I were to compress gas to 40 lbs., a pressure which is used not unfrequently, the theoretical efficiency would be 45 per cent. We actually get something like 24 or 23 per cent.; we know that one-half of the heat is taken away by external cooling. Thus we find a very close coincidence between the calculated efficiency of those engines and that which we actually obtain, only we throw away about one-half of the heat in keeping the cylinder cool enough to permit lubrication. If we compress to 80 lbs. we have a theoretical efficiency of 53 per cent. If we do not compress at all, as Mr. Clerk has told you, we have a theoretical efficiency of only 21 per cent., so that we have it in our power to increase the theoretical efficiency very greatly by increasing the pressure of the gas and air before ignition. I have no doubt that the great gain of efficiency in the Clerk and Otto engines is really due to the fact of the compression; this being done in a workmanlike way and carried to a very considerable point.'

The advantages of compression could not be stated with more clearness and truth.

[1] 'Heat in its Mechanical Applications': *Institution Civil Engineers' Lectures.* Session 1883–84.

In the same year there was published in Paris an able work entitled 'Études sur les Moteurs à Gaz Tonnant,' by Professor Dr. Aimé Witz, of Lille, in which the theoretic efficiencies of the different types of cycle are calculated for a maximum temperature of explosion of 1600° C., and temperature before explosion of 15° C.

He adopts the same classification as the present writer did in 1882, and finds the efficiencies :

$$\begin{aligned}\text{Type } 1 &= 0{\cdot}28\\ \text{Type } 2 &= 0{\cdot}38\\ \text{Type } 3 &= 0{\cdot}44\end{aligned}$$

which are almost identical with the author's figures.

He also arrives at the conclusion that compression is the great source of economy in the modern gas engine. At p. 53 he says : 'I find myself again in agreement with Mr. Dugald Clerk when he affirms that the success of Otto is due to compression alone, and not to the extreme dilution of the explosive mixture in the products of the combustion of a precedent explosion.'

He then proceeds to quote from the present writer's paper, and adheres to the statement that—

'Without compression previous to ignition an engine cannot be produced giving power economically and with small bulk.'

Compression previous to ignition gives two great advantages :

(1) A thermodynamic advantage (improved theory of the cycle) ;

(2) Higher available pressures and smaller cooling surfaces— the joint result being an economy in practice nearly fourfold that of the old non-compression engines.

Mr. Otto's Theory.—Previous to 1882 the nature of the improvement obtained by compression was imperfectly understood, and this notwithstanding the very clear, though qualitative, statements of Schmidt, Million, and Beau de Rochas. An erroneous theory of the cause of the economy of the Otto engine was widely circulated and gained considerable support.

It was enunciated in Mr. Otto's specification of 1876, No. 2081, and it was supported by men so distinguished as Dr. Slaby of Berlin, Professor Dewar of the Royal Institution, the late Sir Frederick Bramwell, and the late Mr. John Imray.

According to Mr. Otto, all gas engines, previous to his patent of 1876, obtained their power from the explosion of a homogeneous charge of gas and air. By the explosion excessive heat was evolved, and the pressures produced rapidly fell away: the excessive heat was rapidly absorbed by the enclosing cold walls.

This caused great loss and gave very wasteful engines. Two methods were open to obtain better economy :

1st, by using a very rapid expansion, so that the heat had but little time to be dissipated ;

2nd, by using slow combustion ; that is, by causing the inflammable mixture to evolve its heat slowly, so that the production of excessive temperatures and pressures was avoided.

By the first method all the heat was supposed to be evolved at once, and a high temperature was produced : by the second method the heat was evolved gradually so as to give a low temperature and pressure which was sustained throughout the stroke, and which was advantageously utilised by the piston while moving at a moderate speed. Mr. Otto states that this gradual evolution of heat may be produced by stratifying the charge of gas and air. Instead of using the homogeneous charge of Lenoir and Hugon, Mr. Otto uses a charge which he states is not homogeneous but heterogeneous. He affirms that his invention lies in the method or process of forming this stratified charge in a gas-engine cylinder, and that in addition to the explosive mixture, there must be present in the cylinder a mass of inert gas which does not burn but which serves to absorb the heat of the explosion and prevent the loss which would otherwise occur by the cooling effect of the cylinder walls.

The 'inert' gas may be either air alone which is capable of supporting combustion, or the products of combustion which are incapable of supporting combustion, or a mixture of both. It is not sufficient that a mere film of this inert gas be present ; there must be what is termed a 'notable' quantity.

Mr. Otto proposes to form this heterogeneous or stratified charge by first drawing into the cylinder a charge of air alone ; and, second, a charge of explosive mixture, or by leaving in the cylinder a sufficient quantity of the products of a previous combustion to form a 'notable' quantity of inert diluent.

The compression space in the Otto engine is supposed to contain a sufficient volume of burned gases to form the inert diluent, so that the whole stroke of the piston is available for taking in the explosive charge.

Suppose the piston to begin its charging stroke : the coal-gas and air mixture flows into the cylinder through the inlet port and mixes to some extent with the inert gas already in the space ; but the mixing is incomplete, and at the piston itself the charge is supposed to consist entirely of exhaust gases. So that, while the charge at the igniting port is readily explosive, that at the piston is not explosive at all, and between the igniting port and the piston the composition of the charge varies from point to point.

This 'arrangement of the gases' is supposed to be retained during compression, and exist at the moment of explosion. The compression

space contains a 'packed charge,' which consists of an explosive mixture at the one end, and between the explosive mixture and the piston a cushion of inert fluid, which is uninflammable and serves the double purpose of relieving the piston from the shock of explosion and absorbing heat which would otherwise be lost by conduction.

By this device heat is gradually evolved. The flame originated in the port burns at first with great energy and spreads from one combustible particle to another, more and more slowly as it approaches the piston, where the particles are dispersed more and more in the inert gas. The mixture is so arranged that this burning lasts throughout the whole stroke, and is complete very shortly before the exhaust valve opens.

The entire cylinder is never completely filled with flame, but the charge at one end has burned out before the flame arrives at the other end.

Dr. Slaby comes forward in support of this hypothesis in an interesting report published as an Appendix to Prof. Fleeming Jenkin's lecture already referred to.

Dr. Slaby states: 'The essence of Otto's invention consists in a definite arrangement of the explosive gaseous mixture, in conjunction with inert gas, so as to suppress explosion (and nevertheless insure ignition).

'At the touch hole, where the igniting flame is applied, lies a strong combustible mixture which ignites with certainty. The flame of this strong charge enters the cylinder like a shot, and during the advance of the piston it effects the combustion of the further layers of dispersed gaseous mixture, whilst the shock is deadened by the cushion of inert gases interposed between the combustible charge and the piston.

'The complete action takes place in a cycle of four piston strokes. The first serves for drawing in the gases in their proper arrangement and mixture; the second compresses the charge; during the third the gases are ignited and expand; and finally, by the fourth the products of combustion are expelled. The essential part of the working is performed by the first of these strokes, by which the charge is drawn in and arranged, first air, then dilute combustible mixture, and finally strong combustible mixture. This arrangement is obtained by the working of the admission slide. Moreover, after discharge of the products of combustion, a portion remains in the clearance space of the cylinder, and this constitutes the inert layer next the piston. By this peculiar arrangement of the gases, the ignition and combustion above described are rendered possible, whilst the products of previous combustion form a cushion, saving the piston from the shock of the

explosion of the strongly combustible mixture at the further end of the cylinder.'

Having stated the essence of Otto's invention, Dr. Slaby proceeds to compare the Otto and Lenoir indicator diagrams, to show that the Otto diagrams prove that the above actions occur in the engine. He finds that the Otto expansion line is somewhat above the adiabatic line, and that the Lenoir expansion line is below it. That is, the Otto diagram gives evidence of heat being added or combustion proceeding in the cylinder during the whole expansion stroke, and the Lenoir diagram gives evidence of loss of heat, not gain, during a similar period. If a mass of expanding gas traces on the diagram the adiabatic line, then it appears as if no loss of heat occurred; but as the temperature of the flame filling the cylinder is known to exceed 1200° C., it must be losing heat to the water jacket. To make the expansion line keep up to the adiabatic a great flow of heat into the gas must be taking place, and as the only source of heat is combustion, it follows that the gas is burning during the expansion period.

Dr. Slaby calculates the proportion of heat evolved by the explosion in the Otto engine as 55 per cent., leaving 45 per cent. to be evolved during expansion.

This he states is due to the portion of the charge which continues to burn after the explosion.

The curve differs from Lenoir's in this, that while in Lenoir's engine *all the heat is evolved at the moment of explosion*, leaving none to be evolved during expansion, in Otto's only a part is evolved at first, and the reserved portion keeps up the temperature during expansion.

He concludes from his experiments that the action of the Otto engine is truly as Mr. Otto states in his specification—explosion is suppressed and a slow evolution of heat is obtained, and this slow evolution of heat is the result of the invention and the cause of the economy of the engine.

In addition to this indirect proof, experiments have been made at Deutz and elsewhere to show directly that stratification has a real existence in the Otto engine.

An Otto engine was constructed, specially fitted with two igniting valves; one valve was placed on the side of the cylinder at the end of the explosion space next the piston, so that it could ignite the gases at the piston; the other valve was the usual one at the end of the cylinder, igniting the gases in the admission port.

Experiments were made to discover if the side valve would fire the mixture at the piston; it was found that it did so. Consecutive ignitions were obtained there.

Diagrams were taken for comparison, with the end and the side valves in alternate action, care being taken to keep the charge in

the same proportions during the trials. It was found that although the side valve ignited as regularly as the end valve, yet the diagrams were different. Instead of the usual rapid ascending explosion line, the explosion took place more slowly, and the maximum pressure was not attained till late in the stroke.

The ignitions were slower from the side valve than from the end valve. If an uninflammable cushion, such as Dr. Slaby so clearly describes, existed at the piston, one would expect that the side valve would fail entirely, but it ignited quite regularly although more slowly than the end valve.

This experiment is considered to prove stratification.

To make stratification visible to the eye, a small glass model was constructed. It consisted of a glass cylinder of about $1\frac{1}{2}$ in. internal diameter, containing a tightly packed piston connected to a crank; the stroke was about 6 ins.; when full back, the piston left a considerable space to represent the explosion space. A brass cover was fitted to the end of the tube, and in it was bored a hole of about $\frac{3}{4}$ in. diameter, representing the admission port; in this hole was screwed a pet cock to which a cigarette was affixed.

On lighting the cigarette and then moving the piston forward by the crank, it was seen that the smoke of the cigarette which passed in did not completely fill the cylinder; the smoke slowly oozed in and left a large clear space between it and the piston. The smoke was supposed to represent the charge of gas and air rushing in, and the clear air behind the piston the cushion which was said to exist in the Otto engine. It was supposed that in the glass cylinder was repeated on a small scale the action of the gases occurring on a larger scale in the Otto engine. In a paper in a German engineering journal, Dr. Slaby recounts this experiment, and lays great weight upon it. He considers that it undoubtedly proves the truth of the Otto theory.

In discussion Mr. John Imray concisely states the Otto position as follows:

'The change which Mr. Otto had introduced, and which rendered the engine a success was this: that instead of burning in the cylinder an explosive mixture of gas and air, he burned it in company with and arranged in a certain way in respect of, a large volume of incombustible gas which was heated by it, and which diminished the speed of combustion.'

And Mr. Bousfield states it in similar terms:

'In the Otto gas engine the charge varied from a charge which was an explosive mixture at the point of ignition to a charge which was merely an inert fluid near the piston. When ignition took place, there was an explosion close to the point of ignition that was gradually

communicated throughout the mass of the cylinder. As the ignition got further away from the primary point of ignition, the rate of transmission became slower, and if the engine were not worked too fast the ignition should gradually catch up the piston during its travel, all the combustible gas being thus consumed. When the engine was worked properly the rate of ignition and the speed of the engine ought to be so timed that the whole of the gaseous contents of the cylinder should have been burned out and have done their work some little time before the exhaust took place, so that their full effect could be seen in the working of the engine. This was the theory of the Otto engine.'

From these quotations it will be seen that Mr. Otto's supporters agree that Mr. Otto has invented a means of suppressing explosion and substituting for explosion a regulated combustion, and that this process is the cause of the economy of the engine. They are agreed that he has succeeded in preventing explosion, and that he does this by arranging or stratifying the charge which is to be used. They consider that engines previous to Mr. Otto's were wasteful because they used a homogeneous and therefore explosive charge, and that Mr. Otto's engine is economical because it uses a heterogeneous or stratified charge, which is consequently non-explosive.

Discussion of Mr. Otto's Theory.—The primary fallacy of Mr. Otto's theory lies in the assumption that previous engines were more explosive than his, and that in previous engines all the heat was evolved at once: as a plain matter of fact this is incorrect. In the Lenoir and Hugon engines, as in all explosive engines, little more than one-half of the total heat is evolved by the explosion,[1] and the portion reserved is evolved during the stroke of the engine.

The following test of a Lenoir engine, made by the author in London, very clearly shows the suppression of heat at first:

Lenoir engine rated at 1 horse-power.
Cylinder $7\frac{1}{8}$ ins. diameter; stroke $11\frac{3}{4}$ ins.
Average revolutions during test, 85 per minute.
Gas consumed in one hour, 86 cubic feet.
With full load, indicated horse-power, 1·17 (average of 9 diagrams).
Gas consumed per indicated horse-power per hour, 73·5 cubic feet.
Maximum temperatures of explosion, 1100° to 1200° C.
Mixture in engine 1 vol. coal gas, 12·5 vols. of air and other gases.
Heat evolved by explosion, 60 per cent. of total heat.

[1] This assumes the working fluid to be air of constant specific heat at high as at low temperatures. See discussion on the working fluid at a later part of this book.

The proportion of the mixture was calculated from the points of cut-off on the diagram and after making allowance for the volume of burned gases in the clearances of the engine. It will be observed that only 60 per cent. of the gas is burned at first, leaving 40 per cent. to be burned during the stroke, and also that the temperature of the explosion never exceeds 1200° C. Now in the Otto engine, according to Thurston, 60 per cent. of the heat is evolved at explosion, and 40 afterwards, and the usual maximum temperature is about 1600° C. So that, so far as the slowness of the explosion is concerned, there is no difference, and in the intensity of the temperature produced the Otto exceeds the Lenoir.

It is difficult to understand how Dr. Slaby could fall into so obvious an error as he did, and suppose that more heat was kept back in the case of the Otto explosion. At the time he wrote his report, accounts of Hirn's, Bunsen's, and Mallard's experiments on explosion were in existence, all of them agreeing on the fact of a large suppression of heat at the maximum temperature of the explosion, although differing in the explanation of the fact.

Hirn even stated that in the Lenoir engine the pressures fell far short of what should be, if all the heat were evolved at once. Yet Dr. Slaby, in the presence of all this definite and carefully ascertained knowledge, is astonished when he finds only 55 per cent. of the total heat evolved by the explosion in the Otto engine, and the only explanation which occurs to him is that of stratification.

If stratification exists at all in the engine, then it produces no measurable change in the explosion; it neither retards the evolution of heat nor does it moderate the temperature.

The explosion and expansion curves are precisely what they would have been with a homogeneous charge.

The mere fact that heat is suppressed in the Otto explosion proves nothing, because a precisely equivalent amount of heat is suppressed in all gaseous explosions, and Dr. Slaby's contention, based upon the supposed peculiarity of the Otto, falls to the ground.

Dr. Slaby has been led into error by the fact that the expansion line of the Lenoir diagram falls below the adiabatic, while the expansion line of the Otto diagram remains slightly above it or upon it. He assumes that in the Lenoir no heat is being added during expansion, whereas just as much heat is being added, or just as much combustion is proceeding during the Lenoir stroke, only the cooling of the cylinder walls is greater, and the heat is abstracted so rapidly that the line falls below the adiabatic. This is due to two causes: (1) the greater proportional cooling-surface exposed by the Lenoir engine, and (2) a longer time of exposure. The absence of compression and the slow piston speed makes the loss greater.

Although quite as much heat is evolved during the stroke, it is overpowered by the greater cooling, and the line falls under the adiabatic. This fall is evidence of greater cooling, not of less evolution of heat.

In a paper,[1] 'Die Verbrennung in der Gasmaschine,' Professor Schöttler makes this explanation of the difference between the lines, and states that 'Whether stratification exists or does not exist in the Otto engine it is unnecessary, and is not the cause of the slow falling of the expansion line.' In all crucial points the Otto theory breaks down, as proved by diagrams taken from his engine.

The explosion is not suppressed; the maximum temperatures produced are not lower than those previously used; the mixture used is not more diluted than in the previous engines, and the intensity of the pressures, as well as the rate of their application, is greater.

The mixture in the engine from Slaby's figures is 1 vol. coal gas to 10·5 vols. of other gases, and from Thurston's figures 1 vol. coal gas to 9·1 vols. of other gases, while Lenoir often used 1 vol. coal gas to 12 of air.

The engine, instead of using a less explosive power than the Lenoir engine, uses one more intensely explosive.

The effect of the reduction of cooling surface and increase of piston velocity is to diminish the loss of heat to the cylinder walls, and the slowly descending line is not the cause of the economy, but is the effect and evidence of it.

Stratification.—The inquiry into the existence or non-existence of stratification in the cylinder has no practical bearing on the question of economy, as the explosion curves act precisely as they would with homogeneous mixtures. Scientifically, however, the question is interesting and will be shortly considered.

The evidence which it is considered proves its existence in the Otto engine is in the author's opinion most unsatisfactory. Dr. Slaby distinctly asserts the existence of an inert stratum next the piston, 'interposed between the combustible charge and the piston,' and Mr. Imray speaks of the 'arrangement of the charge in respect of a large volume of incombustible gases,' and Mr. Bousfield of 'a charge which was merely an inert fluid next the piston.' Yet all the evidence in support of these positive assertions is given by one experiment made with an Otto engine, and one with a small glass model. The evidence given by the experiment on the engine itself, in the author's opinion, disproves stratification in the Otto sense altogether. If the inert stratum next to the piston had any real

[1] *Zeitschrift des Vereines deutscher Ingenieure.* Band xxx., Seite 209.

existence, then the side igniting valve, in the experiment made by Mr. Otto, should not have ignited the mixture at all. The fact that it did ignite regularly and consecutively proved most distinctly that the gas next the piston was not inert but was explosive, and being explosive in itself it could not act as a cushion to absorb heat or shock. That experiment alone settles the question, and proves at once the visionary nature of the cushion of inert gas next the piston.

The fact that the ignitions were slower than those from the end slide does not get rid of the fact that ignition did take place, and to those who understand the sensitive nature of any igniting valve, it will not be difficult to comprehend how small a difference in adjustment will cause late and slow ignitions. At the very utmost the experiment points to a small difference in the dilution of the explosive mixture at the piston and that at the end port.

Experiments made by the author also prove that the mixture in the Otto cylinder is present in explosive proportions close up to the piston. The piston of a $3\frac{1}{2}$ HP Otto engine was bored and fitted with a screw plug, which carried a small spiral of platinum wire in electrical connection with a battery; the platinum spiral projected from the inner surface of the piston by a quarter of an inch. When the engine was running in the usual way, the wire was made incandescent by the battery and the external light was put out. It was proved that by a little care in getting the platinum to a certain temperature the engine worked as usual, igniting regularly and consecutively. The spiral was made just hot enough to ignite when compression was complete, but not hot enough to ignite before compressing. If an incombustible stratum had existed even so close to the piston as $\frac{1}{4}$ in. then the wire should never have been able to ignite the charge at all. If the wire was made too hot, then ignition often took place while the charge was still entering, proving that no stratification existed even while the charge was incomplete. A little consideration of the arrangement of the Otto engine will show that stratification cannot have any existence in it. The end of the combustion space is usually flat, and sometimes the admission port projects slightly into it; the area of the admission port is about $\frac{1}{30}$th of the piston area; accordingly the entering gases flow into the cylinder at a velocity thirty times the piston velocity, or at the Otto piston speed, about 120 miles an hour.

Great commotion inevitably occurs; the entering jet projects itself through the gases right up against the piston, and then returns eddying and whirling till it mixes thoroughly with whatever may be in the cylinder. The mixture becomes practically homogeneous even before compression commences.

Experiments made by the late Dr. John Hopkinson and the author on full-size glass models of the Otto cylinder show this mixing action very beautifully. A $3\frac{1}{2}$ HP Otto cylinder was copied in every proportion in glass, and the valve was so arranged that it passed a charge of smoke at the proper time. The piston was placed at the end of its stroke, leaving the compression space filled with air. When pulled forward the valve opened to a chamber filled with smoke, and the smoke rushed through the port, projected right through the air in the space, struck the piston, and filled the cylinder uniformly, much faster than the eye could follow it. It mixed instantaneously with the air in the cylinder without evincing the slightest tendency to arrange itself in the manner imagined by Mr. Otto. Mr. Otto's experiment with a cigarette and glass cylinder does not, in the most remote degree, imitate the conditions occurring in his engine; the proportions are quite wrong. The model is much too small, and the glass cylinder is too long in proportion to its diameter; then the gases are so badly throttled by passing through the cigarette, that when the piston is moved forward it leaves a partial vacuum behind it, and only a little smoke enters, not nearly enough to follow up the piston, but only sufficient to ooze into the back of the cylinder while the piston moves forward and expands the air which is already in the cylinder. It was easy for Mr. Otto to have copied his cylinder and valve full size and imitated precisely the conditions existing in his engines.

Had he done this he would have proved complete mixing instead of stratification. Why did he refrain from doing this? The question at issue is not, Can stratification be obtained by a specially devised form of apparatus?—no one doubts that it can—but, Does stratification exist in the Otto engine? If it does not exist in the Otto engine then it is perfectly plain that it cannot be the cause of the economy of the motor, and it is quite certain that it cannot exist in the Otto engine. Professor Schöttler, in the paper already referred to, also arrives at the conclusion that stratification has no existence in the Otto engine, and that Mr. Otto's small glass model does not truly represent the actions occurring in the engine.

In all gas engines, when the charge enters the cylinder through a port the residual gases in the port are swept into the cylinder, and while the port itself is filled with gas and air mixture, free from admixture with residual gases, the cylinder contains the gas and air mixture diluted with whatever residual gases exist in the engine which have not been expelled by the piston. The mixture in the port is accordingly stronger and more inflammable than the mixture in the cylinder.

In the Lenoir and Hugon engines this occurred to a marked

extent; in the Hugon engine as much as 30 per cent. of the whole charge consisted of residual gases, and the charge in the cylinder was considerably more dilute than that in the admission port. In the Otto engine this also occurs, but it is not stratification, and it is not a new invention; the cylinder is filled with explosive mixture more dilute than that in the ignition port, but still explosive throughout.

Some space has been devoted here to the discussion of the erroneous theory of the gas engine, as it was very necessary to arrive at correct ideas in order to see the line of true advance. Had the stratification theory been true, advance could not have been made by increasing compression pressures by diminishing the volumes of the compression spaces. This course, however, has been followed, and while in 1876 the compression pressure was only about 30 lbs. per square inch above atmospheric, in 1908 the latest Otto cycle engines utilise pressures as high as 170 lbs. per square inch.

The effect of correct theory upon gas engine advance has been most marked, and facts have proved how much it was necessary to arrive at true conceptions.

Recent experiments have thrown a flood of light upon many of the properties of the working fluid, which will aid materially in further advance, but these matters are fully dealt with later.

Large Gas Engines

The Société Cockerill, of Seraing, continued their work on the large gas engine, and at the end of 1899 they had in operation there a single-cylinder Otto cycle engine, cylinder 51·2 ins. diameter and stroke 55·13 ins. driving a blowing cylinder for a blast furnace. The following particulars will give an idea of its huge dimensions.

Diameter of crank shaft, 18·11 ins.

Piston rod diameter, 9·6 ins.

Height of engine above ground, 13·1 feet; length, 36·1 feet; width, 19·7 feet. Total weight of engine and flywheel without blower, 124·9 tons.

The engine runs at ninety revolutions per minute, gives 600 brake horse-power, and converts 28 per cent. of the heat given to it into indicated work in the cylinder. A number of these engines are shown in their external aspect at fig. 20; their dimensions will be appreciated by observing the workmen shown standing by them.

The engine was fully tested by Professor Hubert, of Liège University, early in 1900, and reported as working well and economically.

It was inspected by the author in 1901, and it was then performing its work with great smoothness and regularity.

About the same time the Gasmotoren Fabrik Deutz took up the subject and built an engine of 1000 HP with four cylinders, each of 33-in. diameter and 39·3-in. stroke, which was set to work at the Hörde Iron Works: it indicated 1200 HP at 135 revolutions per minute. This engine was coupled direct to a dynamo.

The Deutsche-Gas-Kraft Gesellschaft built large Oechelhauser engines on the two-cycle principle, and two 600 HP sets of these engines were also erected at Hörde; each set had two cylinders having two pistons in each: the diameter of the cylinders was 18·9 ins., and the combined stroke of each pair of pistons was 63 ins.

Messrs. Körting, of Hanover, then appeared with a two-cycle engine operating on the Clerk cycle, which speedily took an important position among large gas engines.

Messrs. Crossley Brothers, in England, took up the large gas engine at an early date, and a 400 HP engine by them was at work at Messrs. Brunner, Mond & Co.'s works, Winnington, in 1900; it had two cylinders of 26 ins. diameter and 36 ins. stroke running at 150 revolutions per minute.

The Premier Co., too, have long devoted attention to the large gas engine; they also had a 500 HP engine at work at Winnington in 1900: it had two cylinders $28\frac{1}{2}$ ins. diameter by 30 ins. stroke, and ran at 125 revolutions per minute. It was supplied with a scavenging pump. They now build engines up to 2000 HP.

The National Gas Engine Co., the Hornsby-Stockport Co., Tangyes Ltd., the Campbell Gas Engine Co., Messrs. Fielding & Platt, Ltd., and other English gas-engine builders now construct comparatively large engines, but the largest engines of the double-acting type have been built on the Continent mainly. Most English makers of long experience prefer open-cylinder engines with pistons having no watering arrangements, whereas the standard continental type for high powers is now double-acting four- or two-cycle—single motor cylinders for the two-cycle type, double tandem cylinders for the four-cycle.

English makers have now taken up the manufacture of the leading continental types. Messrs. Richardson, Westgarth & Co., Ltd., the Cockerill engine; Messrs. Mather & Platt, Ltd., the Körting two-cycle; Messrs. Beardmore, the Oechelhauser; the Lilleshall Co., the Nürnberg engines; but in the case of the Cockerill, the Körting, and the Oechelhauser considerable alterations have proved necessary to suit the engines to English use.

Körting engines are built giving 1000 HP per motor cylinder, but Messrs. Ehrhardt & Sehmer claim to exceed this on an Otto cycle engine with an engine giving 1,100 HP per double-acting cylinder. Such an engine with two cranks and two cylinders in tandem to each

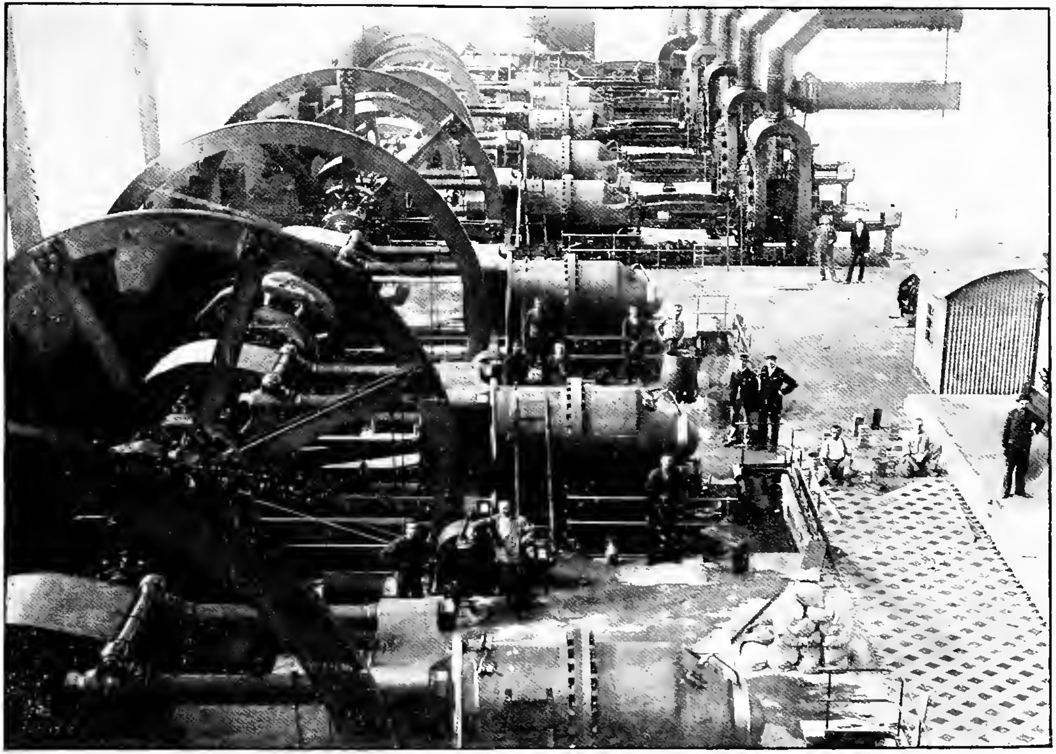


Fig. 20.—Cockerill engines at Differdingen

Six Blowing engines; three Electric-lighting engines. Total power, 5.400 IHP

crank gives a unit of 4,400 HP. Such an engine has cylinders of about $45\frac{1}{4}$ ins. diameter by $51\frac{1}{4}$ ins. stroke, and runs at 94 revolutions per minute. Large engines are undoubtedly making great progress, as will be seen from the following interesting figures by Mr. R. E. Mathot, of Brussels, giving the numbers and horse-power of large gas engines recently manufactured in Europe:

Messrs. Crossley Brothers, Ltd., 57 motors, with an aggregate of 23,660 HP; Messrs. Ehrhardt & Sehmer, 59 motors, total 69,790 HP; the Otto Gasmotoren Fabrik, 82, total 47,400 HP; Gebrüder Körting, 198, total 165,760 HP; Société Alsacienne, 55, total 23,410 HP; Société John Cockerill, 148, total 102,925 HP; Société Suisse, Winterthur, 67, total 8,620 HP; Vereinigten Maschinenfabrik Augsburg and Nürnberg, 215, total 256,240 HP. The mean power of each gas engine made by Messrs. Ehrhardt & Sehmer and the Augsburg and Nürnberg companies is in each case 1,200 HP. It is stated that in one factory there are gas engines representing a total output of 35,000 HP.

These European large gas engines thus give nearly 575,000 HP between them.

In America, too, the large gas engine has made considerable progress. Mr. E. L. Adams estimates that 350,000 HP is now at work or in construction in the United States. The first large engines installed were of the Körting-Clerk type, built by the De La Vergne Co., of New York: sixteen blowing engines of 2000 HP each and eight electric driving engines of 1000 HP each. They were set to work in 1902. Large engines have been built later by the Westinghouse Co., Pittsburg, of the horizontal twin tandem type, having two cranks and four double-acting cylinders on each unit, cylinder 38 ins. diameter by 54 ins. stroke; and the Snow Steam Pump Co. have under construction similar horizontal tandem engines having cylinders of 42 ins. diameter by 54 ins. stroke. The Westinghouse Co. in England has also devoted great attention to the large gas engine, and they have built very interesting multiple-cylinder engines of the single-acting open-trunk type, one of which was at work at the Franco-British Exhibition in the year 1908. It gives 750 HP, is vertical, and has four cranks and eight cylinders, using no watering for the pistons.

Many interesting developments are now in progress; over 2,000,000 HP of smaller gas engines are at work in the world, and certainly over 1,000,000 HP of petrol motors.

The application of large gas engines to marine work, the compounding of the gas engine, and many other matters are being strenuously pursued.

It may be said, indeed, without exaggeration, that the whole world is now alive to the possibilities of the internal-combustion

motor, and that progress will be more and more rapid. These motors have almost fulfilled the expectations of those engineers who—like the author—have devoted a large part of their lives to their study and advancement. They are looking forward now to the completion of the work begun so many years ago, and expect, at no distant date, to find the internal-combustion motor competing with the steam engine even in its latest form, the steam turbine, on sea as vigorously as it does at present on land.

CHAPTER I

THE GAS ENGINE METHOD

GAS ENGINES, while differing widely in theory of action and mechanical construction, possess one feature in common which distinguishes them from other heat engines : that feature is the method of heating the working fluid.

The working fluid is atmospheric air, and the fuel required to heat it is inflammable gas. In all gas engines yet produced, the air and gas are mixed intimately with each other before introduction to the motive cylinder; that is, the working fluid and the fuel to supply it with heat are mixed with each other before the combustion of the fuel.

The fuel, which in the steam and in most hot-air engines is burned in a separate furnace, is, in the gas engine, introduced directly to the motive cylinder and burned there. It is, indeed, part of the working fluid.

This method of heating may be called the gas-engine method, and from it arises at once the great advantages and also the great difficulties of these motors.

Compare first with the steam engine. In it there exist two great causes of loss : water is converted into steam, absorbing a great amount of heat in passing from the liquid to the gaseous state ; after it has been used in the engine it is rejected into the atmosphere or the condenser, still existing as steam. The heat necessary to convert it from the liquid to the gas is consequently in most part rejected with it. Loss, occurring in this way, would be small if high temperatures could be used ; but this is the point where steam fails. High temperatures cannot be obtained without pressure so great as to be quite unmanageable. The attempt to obtain high temperatures by superheating has often been made, but without any substantial success. Although the difficulty of excessive pressure is avoided, another set of troubles is introduced. All the heat to be given to the gaseous steam must pass through the iron plates forming the boiler or superheater, which plates will only stand a comparatively low temperature, certainly not exceeding that of a low red heat, or about 600° to 700° C. Steam, being a gas, is much more difficult

to heat than water; it follows that even these temperatures cannot be attained without enormous addition to the heating surface. The difficulties of making a workable engine using high-temperature steam are so great that even so distinguished an engineer and physicist as the late Sir C. W. Siemens failed in his attempts, which extended over many years. It may be taken, then, that low temperature is the natural and unavoidable accompaniment of the steam method, arising from the necessary change of the physical state of the working fluid and the limited temperature which iron will safely bear. The originators of the science of thermodynamics have long taught that the maximum efficiency of a heat engine is obtained when there is the maximum difference between the highest and lowest temperatures of the working fluid. So long ago as 1854 Professor Rankine read a paper before the British Association, 'On the means of realising the advantages of the Air Engine,' in which he expresses his belief that such engines will be found to be the most economical means of developing motive power by the agency of heat. In this opinion he stood by no means alone. Engineers so able as Stirling, Ericsson, and Siemens, physicists so distinguished as Dr. Joule and Sir Wm. Thomson, devoted much energy and study to their practice and theory. Notwithstanding all their efforts, aided by a host of less able inventors, the difficulties proved too formidable; and although more than fifty years have now passed since Rankine announced his belief, the hot-air engine proper has made no real advance. Similar causes to those acting in the steam engine impose a limit here. It is true the complication of changing physical state is avoided, but the limited resistance of iron to heat acts as powerfully as ever. Air is much more difficult to heat than water, and therefore requires a much larger surface per unit of heat absorbed. In the larger hot-air engines, accordingly, the furnaces and heating surfaces gave great trouble. Very low maximum temperatures were attained in practice. In a Stirling engine giving out thirty-seven brake horse-power, the maximum temperature was only 343° C.; in the engines of the ship 'Ericsson,' the maximum was only about 212° C., according to Rankine, the indicated power being about 300 horses. These figures show that the heating surfaces were insufficient, as in both cases the furnaces were pushed to heat the metal to a good red. A method of internal firing was proposed, first by Sir George Cayley and afterwards carried out with some success by others; the furnace was contained in a completely closed vessel, and the air to be heated was forced through it before passing to the motor cylinder. The plan gave better results, but the temperature of 700° C. was still the limit, as the strength of the iron reservoir had to be considered, and the hot gases had to pass through valves. Wenham's engine, described in a paper read before the

Institution of Mechanical Engineers in 1873, is a good example of this class. In it the highest temperature of the working fluid, as measured by a pyrometer, was 608° C.; higher temperatures could easily have been got, but the safety of the engine did not permit it. Professor Rankine in his work on the steam engine has very fully discussed the disadvantages arising from low maximum temperatures. He calculates that in a perfect air engine without regenerator an average pressure of 8·3 lbs. per square inch would only be attained with a maximum of 216·6 lbs. per square inch, thus necessitating great strength of cylinder and working parts for a very small return in effective power. In the 'Ericsson,' the average effective pressure was less than this, being only about 2 lbs. per square inch; it had four air cylinders each of 14 feet diameter, and only indicated 300 horse-power. Stirling's motor cylinder did not give a true idea of the bulk of the engine, as the real air-displacer was separate. Even with Wenham's machine the bulk was excessive, an engine of 24 ins. diameter cylinder and 12 ins. stroke giving 4 horse-power.

Those facts sufficiently illustrate the practical difficulties which prevented the development of the hot-air engine proper. All flow from the method of heating. Low temperature is necessary to secure durability of the iron.

All hot-air engines are, therefore, very large and very heavy for the power they are capable of exerting.

The friction of the parts is so great that although the theoretical efficiency of the working fluid is higher than in the best steam engines, the practical efficiency or result per horse available for external work is not nearly so great. The best result ever claimed for Stirling's engine is 2·7 lbs. of coal per brake horse-power per hour, probably under the truth, but even allowing it, a first-class steam engine of to-day will do much better. According to Professor Norton, the engines of the 'Ericsson' used 1·87 lb. of anthracite per indicated horse-power per hour; but the friction must have been enormous. Compared with the steam engine, the practical disadvantages of the hot-air engine are much greater than its advantage of theory. Owing to the great inferiority of air to boiling water as a medium for the convection of heat, the efficiency of the furnace is much lower; owing to the high maximum and low available pressure, the friction is much greater—which disadvantages in practice more than extinguish the higher theoretical efficiency.

The gas engine method of heating by combustion or explosion at once disposes of those troubles; it not only widens the limits of the temperatures at command almost indefinitely, but the causes of failure with the old method become the very causes of success with the new.

The difficulty of heating even the greatest masses of air is quite abolished. The rapidly moving flash of chemical action makes it easy to heat any mass, however great, in a minute fraction of a second; when once heated the comparatively gradual convection makes the cooling a very slow matter. The conductivity of air for heat is but slight, and both losing and receiving heat from enclosing walls are carried on by the process of convection, the larger the mass of air the smaller the cooling surface relatively. Therefore the larger the volumes of air used, the more economical the new method, the more difficult the old. The low conductivity for heat, the cause of great trouble in hot-air machines, becomes the unexpected cause of economy in gas engines. If air were a rapid carrier of heat, cold cylinder gas engines would be impossible. The loss to the sides of the enclosing cylinders would be so great that but little useful effect could be obtained. Even as it is, present loss from this cause is sufficiently heavy. In the earlier engines as much as three-fourths of the whole heat of the combustion was lost in this way; in the best modern engines so much as one-quarter is still lost.

A little consideration of what is occurring in the gas-engine cylinder at each explosion will show that this is not surprising. Platinum, the most infusible of metals, melts at about 1700° C.; the ordinary temperature of cast iron flowing from a cupola is about 1200° C.; a temperature very usual in a gas-engine cylinder is 1600° C., a dazzling white-heat. The whole of the gases filling the cylinder are at this high temperature. If one could see the interior it would appear to be filled with a blinding glare of light. This experiment the writer has tried by means of a small aperture covered with a heavy glass plate, carefully protected from the heat of the explosion by a long cold tube. On looking through this window while the engine is at work, a continuous glare of white light is observed. A look into the interior of a boiler furnace gives a good notion of the flame filling the cylinder of a gas engine.

At first sight it seems strange that such temperature can be used with impunity in a working cylinder; here the convenience of the method becomes evident. The heating being quite independent of the temperature of the walls of the cylinder, by the use of a water jacket they can be kept at any desired temperature. The same property of rapid convection of heat, so useful for generating steam from water, is essential in the gas engine to keep the rubbing surfaces at a reasonable working temperature. In this there is no difficulty, and notwithstanding the high temperature of the gases, the external surface of metal itself never exceeds the boiling-point of water.

So good a result cannot of course be obtained without careful

proportioning of the cooling surfaces for the amount of heat to be carried away; in all modern engines this is carefully attended to, with the gratifying result that the cylinders take and retain a polished surface for years of work just as in a good steam engine.

The gas engine method gives the advantage of higher temperature of working fluid than is attainable in any other heat engine, and at the same time the working cylinder metal may be kept as cool as in the steam engine. It also allows of any desired rate of heating the working fluid in any required volumes.

In consequence of high temperatures the available pressures are high, and therefore the bulk of the engine is small for the power obtained.

It realises all the thermodynamic advantages claimed for the hot-air engine without sacrificing the high available pressures and rapid rate of the generation of power which is the characteristic of the steam engine.

For rapid convection of heat existing in the steam boiler is substituted the still more rapid heating by explosion or combustion, a rapidity so superior that the power is generated for each stroke separately as required, there being no necessity to collect a great magazine of energy.

The only item to the debtor side of the gas engine account is the flow of heat through the cylinder walls, which disadvantage is far more than paid for by the advantages.

CHAPTER II

GAS ENGINES CLASSIFIED

ALTHOUGH the gas-engine patents now in existence number many thousands, the essential differences between the inventions are not great. In their working process they may be divided into a few well-defined types :

1. Engines igniting at constant volume, but without previous compression.

2. Engines igniting at constant pressure, with previous compression.

3. Engines igniting at constant volume, with previous compression.

THE FIRST TYPE is the simplest in idea ; it is the most apparent method of obtaining power from an explosion.

In it the engine draws into its cylinder gas and air at atmospheric pressure, for a part of its stroke, in proportions suitable for explosion ; then a valve closes the cylinder, and the mixture is ignited. The pressure produced pushes forward the piston for the remainder of its travel, and upon the return stroke the products of the combustion are expelled exactly as the exhaust of a steam engine. By repeating the same process on the other side of the piston, a kind of double-acting engine is obtained. It is not truly double-acting, as the motive impulse is not applied during the whole stroke, but only during that portion of it left free after performing the necessary function of charging with the explosive mixture.

The working cycle of the engine consists of four operations :

1. Charging the cylinder with explosive mixture.
2. Exploding the charge.
3. Expanding after explosion.
4. Expelling the burned gases.

To carry it out in a perfect manner, the mechanism must be so arranged that during the charging the pressure of the gases in the cylinder does not fall below atmosphere ; there must be no throttling of the entering gases. The cut-off and the explosion must be absolutely simultaneous and also instantaneous, so that the heat may be applied without change of volume, and thereby

produce the highest pressure which the mixture used is capable of giving. The expansion will be carried far enough to reduce the pressure of the explosion to atmosphere; and the exhaust stroke will be accomplished without back pressure. The charge in entering must not be heated by the walls of the cylinder, but should remain at the temperature of the atmosphere till the very moment previous to ignition. At the same time, the cylinder should not cool the gases after the explosion and no heat should disappear except through expansion doing work.

Although all these conditions are necessary to the perfect cycle, it is evident that no actual engine is capable of combining them. Some throttling at the admission of the mixture, and a little back pressure during the exhausting are unavoidable; some time must elapse between the closing of the inlet valve and the explosion, in addition to the time taken by the explosion itself. Heat will be communicated to the entering gases and lost by the exploded gases to the walls of the cylinder.

The actual diagram taken from an engine will therefore differ considerably from the theoretical one.

The theoretical conditions are to a great extent contradictory.

The idea of the type, however, is easily comprehensible, and evidently suggested by the common knowledge of the destructive effect of accidental coal-gas explosions which occurred soon after the introduction of gas into general use. 'The power is there, let us use it like steam in the cylinder of a steam engine,' said the early inventors.

The two most successful engines of this type were Lenoir's and, later, Hugon's.

THE SECOND TYPE is not so simple in its main idea, and required much greater knowledge of detail, both mechanical and theoretical. As a hot-air engine its theory was originally proposed by Sir George Cayley, and, later, by Dr. Joule and Sir Wm. Thomson. As a hot-air engine it failed for the reasons discussed in the previous chapter.

In it the engine is provided with two cylinders of unequal capacity; the smaller serves as a pump for receiving the charge and compressing it, the larger is the motor cylinder, in which the charge is expanded during ignition and subsequent to it.

The pump piston, in moving forward, takes in the charge at atmospheric pressure, in returning compresses it into an intermediate receiver, from which it passes into the motor cylinder in a compressed state. A contrivance similar to the wire gauze in a Davy lamp commands the passage between the receiver and the cylinder, and permits the mixture to be ignited on the cylinder side as it flows in without the flame passing back into the receiver.

The motor cylinder thus receives its working fluid in the state of flame, at a pressure equal to, but never greater than, the pressure of compression. At the proper time the valve between the motor and the receiver is shut, and the piston expands the ignited gases till it reaches the end of its stroke, when the exhaust valve is opened, and the return expels the burned gases.

The ignition here does not increase the pressure, but increases the volume. The pump, say, puts one volume or cubic foot into the receiver ; the flame causes it to expand while entering the cylinder to two cubic feet. It does the work of two cubic feet in the motor cylinder, so that, though there is no increase of pressure, there is nevertheless an excess of power over that spent in compressing.

In the first type of engine the heat is given to the working fluid at constant volume, in the second type the heat is given to the working fluid at constant pressure during change of volume.

The working cycle of the engine consists of five operations :

1. Charging the pump cylinder with gas and air mixture.

2. Compressing the charge into an intermediate receiver.

3. Admitting the charge to the motor cylinder in the state of flame, at the pressure of compression.

4. Expanding after admission.

5. Expelling the burned gases.

To carry out the process perfectly the following conditions would be required :

No throttling during admission of the charge to the pump.

No heating of the charge as it enters the pump from the atmosphere.

No loss of the heat of compression to the pump and receiver walls.

No throttling as the charge enters the motor cylinder from the receiver.

No loss of heat by the flame to the sides of the motor cylinder and piston.

And last, No back pressure during the exhaust stroke.

The exhaust gases also must be completely expelled by the motor piston ; that is, the motor cylinder should have no clearance.

The requirements of this type, although sufficiently numerous and exacting, are not so contradictory among themselves as in the first.

Although every engine of the kind yet made fails to fulfil them, it is quite possible that a machine very closely approximating may be yet constructed.

The most successful engines of this kind have been Brayton's and Simon's, the first an American invention, and the second an

English adaptation of it. Sir C. W. Siemens proposed such an engine in 1861, but does not seem to have been successful in carrying it out. In 1860 it was also proposed by F. Million, but without a sufficient understanding of the mechanical detail necessary for a working machine.

Brayton's engine was made in considerable numbers in America, and was applied by him to drive a good-sized launch, petroleum being used as the fuel instead of gas. It was exhibited at the Centennial Exhibition in Philadelphia; and at the Paris Exhibition of 1878 by Simon.

THE THIRD TYPE is the best kind of compression engine yet introduced; by far the largest number of gas engines in everyday use throughout the world are made in accordance with its requirements. In theory it is more easily understood as requiring two cylinders, compression and power.

The leading idea, compression and ignition at constant volume, was first proposed by Barnett in 1838, then by Schmidt in more general terms, very fully by Beau de Rochas in 1860 and also by F. Million in the same year. Otto, however, was the first successfully to apply it, which he did in 1876.

The compression cylinder may be supposed to take in the charge of gas and air at atmospheric temperature and pressure; compress it into a receiver from which the motor cylinder is supplied; the motor piston to take in its charge from the reservoir in a compressed state; and then communication to be cut off and the compressed charge ignited.

Here ignition is supposed to occur at constant volume, that is, the whole volume of mixture is first introduced and then fired; the pressure therefore increases. The power is obtained by igniting while the volume remains stationary and the pressure increases.

Under the pressure so produced, the piston completes its stroke, and upon the return stroke the products of the combustion are expelled.

In this case the working cycle of the engine consists of six operations:

1. Charging the pump cylinder with gas and air mixture.
2. Compressing the charge into an intermediate receiver.
3. Admitting the charge to the motor cylinder under compression.
4. Igniting the mixture after admission to the motor.
5. Expanding the hot gases after ignition.
6. Expelling the burned gases.

To carry out the process perfectly, similar conditions are necessary to those in the second type. But the conditions are more

contradictory. The gases entering the cylinder under pressure must not be heated by its walls; no heat should be added till the ignition; then, after ignition, the gases must not lose heat to the cylinder—conditions which it is impossible for the same cylinder to fulfil simultaneously.

In the engines constructed the receiver is dispensed with, for reasons which will be explained in discussing the practical difficulties of construction; but this does not in any way modify the theory, which shall first be discussed.

The most considerably used engines of this kind are of the Otto or the Clerk type. In neither of these types does any part of the working cycle require either the heating or the cooling of the working fluid by the relatively slow processes of convection and conduction.

Heating is accomplished by the rapid method of explosion or, if the term be preferred, combustion, and for the cooling necessary in all heat engines is substituted the complete rejection of the working fluid with the heat it contains and its replacement by a fresh portion taken from the atmosphere at the atmospheric temperature, which is the lower limit of the engines.

This is the reason why those cycles can be repeated with almost indefinite rapidity, and why gas engines can be run at speeds equal to steam engines, while the old hot-air engines could not be run fast, because of the very slow rate at which air could be heated and cooled by contact.

There still remains one important type of gas engine not included in this classification; in it part of the efficiency is dependent on cooling by contact, and consequently only a slow rate of working stroke can be obtained. It is the kind of engine known as the free piston or atmospheric gas engine. It may be regarded as a modification of the first type. The first part of its action is precisely similar, the latter part differs considerably.

It may be called *Type* 1 A. In it the piston moves forward, taking in its charge of gas and air from the atmosphere at the atmospheric pressure and temperature. When cut off it is ignited instantaneously, the volume being constant and the pressure increasing; the piston is not connected directly to the motor shaft, but is free to move under the pressure of the explosion, like the ball in a cannon. It is shot forward in the cylinder (which is made purposely very long); the energy of the explosion gives the piston velocity; it therefore continues to move considerably after the pressure has fallen by expansion to atmosphere; a partial vacuum forms under the piston till its whole energy of motion is absorbed in doing work upon the exterior air. It then stops, and the external pressure causes it to perform its instroke, during which a clutch arrangement yokes it to the motor

shaft, giving the shaft an impulse. The explosion is made to give its equivalent in work upon the external air, in forming a vacuum in fact; the vacuum is increased by the cooling of the hot gases during the return of the piston. The piston proceeds completely to the bottom of the cylinder, expelling the products of combustion. So far as the working fluid of the engine is concerned the cycle consists of five operations:

1. Charging the cylinder with explosive mixture.
2. Exploding the charge.
3. Expanding after explosion.
4. Compressing the burned gases after some cooling.
5. Expelling the burned gases.

To carry it out perfectly, in addition to the requirements of the first type, the expansion should be carried far enough to lower the temperature of the working fluid to the temperature of the atmosphere, and the compression to atmospheric pressure again should be conducted at that temperature; that is, the compression line should be an isothermal.

This kind of engine was proposed first by Barsanti and Matteucci in 1854, by F. H. Wenham in 1864, and then by Otto and Langen in 1866. The last-named inventors were successful in overcoming the practical difficulties, and many engines were made and sold by them. Their engine, although cumbrous and noisy, was a good and economical worker. The next best known engine of the kind was Gillies's, of which a considerable number were constructed and sold.

CHAPTER III

THERMODYNAMICS OF THE INTERNAL-COMBUSTION ENGINE CONSIDERED AS AN AIR ENGINE

BEGINNING with Professor Rankine, able writers have so fully treated the thermodynamics of the air engine that but little can be added to the knowledge of the subject now in existence. The gas engine method of heating, however, introduces limits of temperature so extended and cycles of action so different from those possible in the air engine proper, that something remains to be done in applying the existing data. So far as the author is aware, this had only been previously attempted by three writers prior to 1885—Professor R. Schöttler, Dr. A. Witz, and himself.

Before proceeding with the special consideration of the subject, it is advisable for the sake of completeness to state briefly the general laws. In doing so Rankine will be followed as closely as possible.

THERMODYNAMICS DEFINED

'It is a matter of ordinary observation that heat, by expanding bodies, is a source of mechanical energy, and conversely, that mechanical energy, being expended either in compressing bodies or in friction, is a source of heat.

'The reduction of the laws according to which such phenomena take place to a physical theory or connected system of principles constitutes what is called the science of thermodynamics.'

FIRST LAW OF THERMODYNAMICS

Heat and mechanical energy are mutually convertible, and heat requires for its production, and produces by its disappearance, mechanical energy in the proportion of 1,390 foot-pounds for each Centigrade heat unit, a heat unit being the amount of heat necessary to heat one pound weight of water through 1° C. This is Joule's law, having been first determined by him in 1843. It holds with equal truth for other forms of energy, and is a general statement of the great truth that in the universe energy is as incapable of creation

or destruction as matter. Energy may change its form indefinitely while passing from a higher to a lower level, but it can neither be created nor destroyed. The energy of outward and visible movement of matter may be arrested and caused to disappear as movement of the whole mass in one direction, but its equivalent reappears as internal movement or agitation of the particles or molecules composing the body. Energy assumes many forms, but the sum of all remains a constant quantity, incapable of change of quantity, but capable of disappearing in one form and reappearing in another.

Second Law of Thermodynamics

Although heat and work are mutually convertible and in definite and invariable proportions, yet no conceivable heat engine is able to convert all the heat given to it into work.

Apart altogether from practical limitations, a certain portion of the heat must be passed from the hot body to the cold body in order that the remainder may assume the form of mechanical energy. To get a continuous supply of mechanical energy from heat depends upon getting a continuous supply of hot and cold substances : it is by the alternate expansion and contraction of some substance, usually steam or air, that heat is converted into mechanical energy.

Perfect heat engines are ideal conceptions of machines which are practically impossible, but whose operations are so arranged that, if possible, they would convert the greatest conceivable proportion of the heat given to them into mechanical work.

Efficiency.—The efficiency of a heat engine is the ratio of the heat converted into mechanical work to the total amount of heat which enters the engine.

In this work the word *Efficiency*, when used without qualification, bears this meaning only.

The efficiency of a perfect heat engine depends upon two things alone : these are, the temperature of the source of heat and the temperature of the source of cold (allowing the expression). The greater the difference between these temperatures the greater the efficiency. That is, the greater will be the proportion of the total heat converted into mechanical energy, and the smaller the proportion of the total heat which necessarily passes by conduction from the hot to the cold body.

Properties of Gases.—Gases are the most suitable bodies for use in heat engines ; they are almost perfectly elastic, and they expand largely under the influence of heat.

A gas is said to be perfect when it completely obeys two laws :

1. Boyle's law.
2. Charles's law.

Boyle's Law.—Suppose unit volume of gas to be contained in a cylinder fitted with a piston which is perfectly tight at unit pressure. Suppose the temperature to be kept perfectly constant. Then, according to Boyle's law, however the volume may be changed by moving the piston, the pressure is always inversely proportional to volume—that is, if the volume becomes two, the pressure becomes one-half; volume becomes three, pressure becomes one-third.

The product of pressure and volume is always constant.

Denoting pressure by p, and volume by v,

$$\text{Boyle's law is, } p\,v = \text{constant.}$$

Charles's Law.—If a gas kept at constant volume is heated, the pressure increases. If a gas is kept behind a piston which moves without friction so that the pressure upon the gas is always constant, the heat applied will cause it to expand.

One volume of gas at 0° C., if heated through 1° C. will expand $\frac{1}{273}$, and become $1\frac{1}{273}$ volume, if the pressure is constant. If the volume is constant, then its pressure will increase by $\frac{1}{273}$, that is, its pressure will become $1\frac{1}{273}$ of the original. In the same way if cooled 1° C. below 0° C., it will contract or diminish in pressure by $\frac{1}{273}$, its volume or pressure becoming $\frac{272}{273}$ of what it is at 0° C.

For every degree of heat or cold above or below 0° C. a perfect gas expands or contracts by $\frac{1}{273}$ of its volume at 0° C.

From this it is evident that a perfect gas, if cooled to 273 Centigrade degrees below 0° C. will have neither volume nor pressure.

This originally gave rise to the conception of absolute zero of temperature. The absolute temperature of a body is ordinary temperature in degrees Centigrade + 273, just as the absolute pressure of any gas is its pressure above atmosphere plus atmospheric pressure. The absolute temperature of a body is its temperature above Centigrade zero + 273.

The pressure or volume of a gas is therefore directly proportional to its absolute temperature.

If p=pressure for absolute temperature t, and p^1 pressure for t' temperature, also absolute,

$$\text{then } \frac{p}{p^1} = \frac{t}{t^1};$$

or if v be the volume at absolute temperature t and v^1 at t^1,

$$\text{then } \frac{v}{v^1} = \frac{t}{t^1}.$$

The Second Law (quantitative).—If heat be supplied to a perfect heat engine at the absolute temperature T, and the absolute tem-

perature of the source of cold is T^1, then the efficiency of that engine is, denoting it by E,

$$E = \frac{T - T^1}{T} = 1 - \frac{T^1}{T}.$$

It is unity minus the lower temperature divided by the upper temperature. The efficiency is greater or less as the fraction $\frac{T^1}{T}$ is less or greater. This fraction may be diminished either by reducing T or by increasing T. The lowest available temperature is not capable of great variation, being in our climate about 290° C. absolute. It therefore follows that efficiency could only be increased by increasing T.

Suppose $T^1 = 290°$ absolute and $T = 580°$ absolute.

$$\text{Then} \quad E = 1 - \tfrac{290}{580} = 1 - \tfrac{1}{2} = 0{\cdot}5.$$

Suppose $T^1 = 290°$, and $T = 1450°$, a temperature common in gas engines, then

$$E = 1 - \tfrac{290}{1450} = 1 - \tfrac{1}{5} = 0{\cdot}8.$$

The efficiency increases with increase of the maximum temperature. The second law, in its quantitative form, is the statement of the efficiency of any perfect heat engine in terms of absolute temperatures of the source of heat and the source of cold.

Thermal Lines.—If a volume of air is contained in a cylinder having a piston and fitted with an indicator, the piston, if moved to and fro, will alternately compress and expand the air, and the indicator pencil will trace a line or lines upon the card, which lines register the changes of pressure and volume occurring in the cylinder. If the piston is perfectly free from leakage, and it be supposed that the temperature of the air is kept quite constant, then the line so traced is called an *Isothermal line,* and the pressure at any point when multiplied by the volume is a constant according to Boyle's law,

$$pv = \text{a constant.}$$

If, however, the piston is moved in very rapidly, the air will not remain at constant temperature, but the temperature will increase because work has been done upon the air, and the heat has no time to escape by conduction. If no heat whatever is lost by any cause, the line will be traced over and over again by the indicator pencil, the cooling by expansion doing work precisely equalling the heating by compression. This is the line of no transmission of heat, therefore, known as *Adiabatic.* Fig. 21 shows these two lines for air starting from atmospheric pressure and temperature.

The pressures at different points of the curve are related by the equation

$$pv^{\gamma} = \text{constant}.$$

The pressure when multiplied by the volume raised to the γ power is always constant.

The power γ is the ratio between the specific heat of the air at constant pressure and its specific heat at constant volume. According to Rankine

$$\gamma = 1{\cdot}408 \text{ for air}.$$

Imperfect Heat Engines.—For a complete description of the working cycle of perfect heat engines, the reader is referred to works upon the

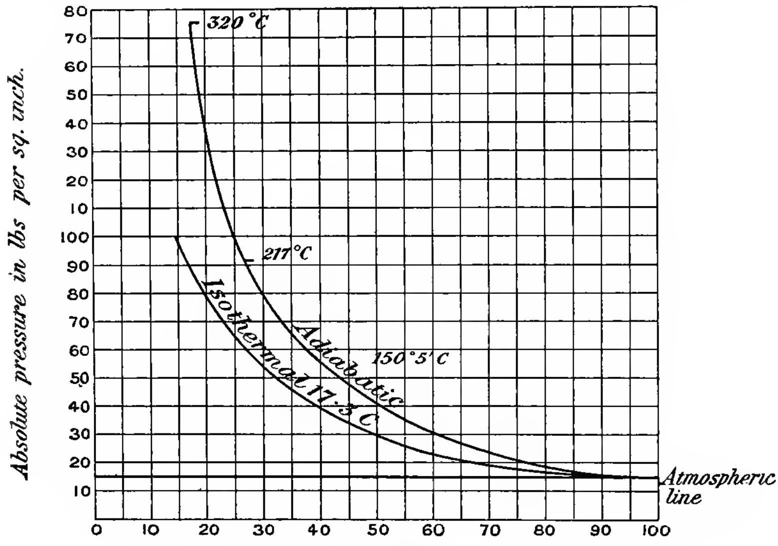


FIG. 21.—Compression lines for air (dry), Adiabatic and Isothermal

steam engine, which contain the fullest possible details both of reasoning and of results.

The working cycles of practicable heat engines are always imperfect, that is, the operations are such that, although perfectly carried out, the maximum efficiency possible by the second law of thermodynamics could not be attained by them. Each cycle has a maximum efficiency peculiar to itself, which is invariably less than $\frac{T - T^1}{T}$, but which does not necessarily vary with T and T^1.

It does not always follow that increase of the higher temperature causes increase of efficiency; conversely, it does not always

follow that diminution of the upper temperature causes diminution of efficiency. Under some circumstances, indeed, the opposite effect is produced—increase of the upper temperature diminishes efficiency, while its diminution increases it, of course within certain limits.

All the gas-engine cycles described in the previous chapter are imperfect in this sense, but all are practicable. It follows that if any one of them gives a higher efficiency than another in theory, it will also do so in practice, provided the practical losses do not increase with improved theory.

It is necessary before discussing the practical losses to see how the cycles compare with each other, if each be perfectly carried out. The results obtained can then be modified by examination of the way in which unavoidable practical losses affect each cycle.

Efficiency Formulæ

If H is the quantity of heat given to an engine, and H^1 the amount of heat discharged by it after performing work, then, the portion which has disappeared in performing work is $H - H^1$, supposing no loss of heat by conduction or other cause, and the efficiency of the engine is

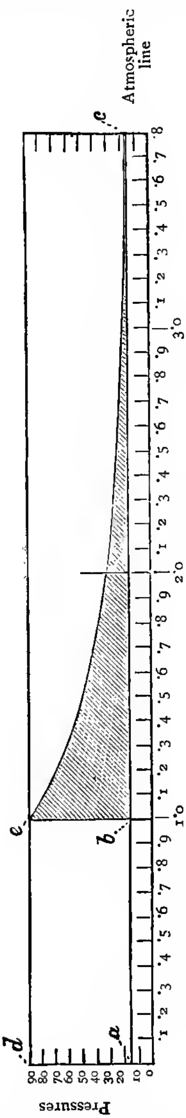
$$E = \frac{H - H^1}{H}.$$

Type I.—A perfect indicator diagram of an engine of this kind is shown at fig. 22 : the line *a b c* is the atmospheric line, representing volume swept by the piston, the line *a d* is the line of pressures. From *a* to *b* the piston moves forward, taking in its charge, at atmospheric temperature and pressure; at *b* communication is instantaneously cut off, and heat instantaneously supplied, raising the temperature to the maximum, before the movement of the piston has time to change the volume. From *e*, the point of maximum temperature and pressure, the gases expand without loss of heat, the temperature only falling by reason of work performed till the pressure again reaches atmosphere. The curve *e c* is therefore adiabatic. In all cases let

t be the initial temperature of the air in absolute degrees Centigrade.
T the absolute temperature after explosion or heating.
T^1 the absolute temperature of the gases after adiabatic expansion.
p the atmospheric pressure.
P_0 the absolute pressure of the explosion.
v_o the volume at atmospheric temperature and pressure.
v the volume at the termination of adiabatic expansion.

In the particular case of diagram fig. 22, where the expansion is

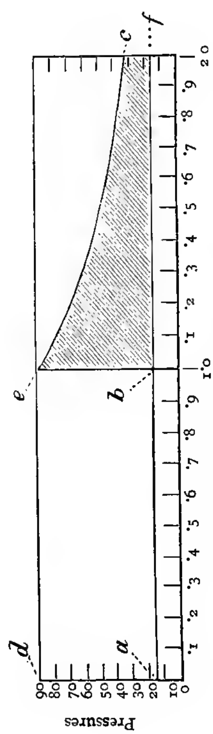
continued to the atmospheric line, the formula expressing the efficiency is very simple. Calling K_v the specific heat of air at constant volume,



VOLUMES

t absolute temp. °C. at b — p absolute pressure at b
T " " e — p_0 " " c
T' " " c — P0 " " e
v_0 volume at b
v " " c

FIG. 22.—Type I. Perfect diagram. Complete expansion



VOLUMES

t absolute temp. °C. at b — p absolute pressure at b
p_0 " " c
T " " e — P0 " " e
T' " " c — v_0 volume at b
t' " " f — v " " c

FIG. 23.—Type I. Perfect diagram. Incomplete expansion

and K_p the specific heat at constant pressure, then the heat supplied to the engine is

$$H = K_v (T - t),$$

and the heat discharged from it is

$$H^1 = K_p\,(T^1 - t)\,;$$

therefore efficiency is

$$E = \frac{K_v\,(T - t) - K_p\,(T^1 - t)}{K_v\,(T - t)},$$

and $$\frac{K_p}{K_v} = \gamma$$

therefore $$E = 1 - \gamma\left(\frac{T^1 - t}{T - t}\right) \tag{1}$$

It is evident that for every value of T there is a corresponding value of T^1, which increases with the increase of T. If T is known in terms of T, then the calculation of efficiency is very rapid, as all that is required is a knowledge of the maximum temperature of the explosion to calculate the efficiency of an engine using that maximum temperature, and perfectly fulfilling this cycle.

For any adiabatic curve, the pressure multiplied by volume which has been raised to the power γ is a constant; therefore

$$P_o\, v_o^{\gamma} = p_o v^{\gamma} \text{ (see diagram, fig. 22),} \tag{a}$$

and $$\frac{T}{t} = \frac{P_o}{p} \text{ which, as } p = p_o, \text{ is the same as } \frac{P_o}{p_o}\,;$$

also $$\frac{v}{v_o} = \frac{T^1}{t}.$$

$\therefore$ in equation (a) T may be substituted for P_o, t for p_o, t for v_o, and T^1 for v, giving

$$T\, t^{\gamma} = t\, T^{1\gamma}$$

$$T^1 = t\left(\frac{T}{t}\right)^{\frac{1}{\gamma}} \tag{2}$$

In most engines of this type the expansion is not great enough to reduce the pressure to atmosphere before opening the exhaust valve; it is therefore necessary to give formulæ where the best condition is not carried out. Fig. 23 is a diagram of a case of this kind.

The pressure at the termination of the stroke has fallen to p_o, and the temperature to T^1. The heat supplied to the engine is the same as in the first case

$$H = K_v\,(T - t).$$

The heat discharged by it cannot be so simply expressed. Suppose the hot gases at the pressure p_o to be allowed to cool by contact with the sides of the cylinder at constant volume till the atmospheric pressure p is reached, then the temperature

$$t^1 = T^1\,\frac{p}{p_o},$$

or in terms of volume and t $$t^1 = \frac{v}{v_o} t,$$

and the heat lost is $$K_v (T^1 - t^1).$$

The heat to be still abstracted before the air returns to its original condition at t, and pressure p is

$$K_p (t^1 - t).$$

Total heat discharged by exhaust, therefore,

$$H^1 = K_v (T^1 - t^1) + K_p (t^1 - t).$$

The efficiency consequently is

$$E = \frac{K_v (T - t) - \{K_v (T^1 - t^1) + K_p (t^1 - t)\}}{K_v (T - t)}$$

$$= 1 - \frac{(T^1 - t^1) + \gamma (t^1 - t)}{T - t} \qquad (3)$$

In this case there is no fixed relationship between T the temperature of the explosion, and T^1 the temperature of the gases at the termination of adiabatic expansion. As the expansion is more or less complete, so do T^1 and t^1 change. In no case, however, can the efficiency be so great as that in the first case.

Type 2.—A perfect indicated diagram of an engine of this type is shown at fig. 24. Although the cycle requires two cylinders, producing two diagrams, they are better compared when superposed. The whole diagram may be supposed to come from the motor cylinder, the shaded portion of it representing the available work of the cycle, and the unshaded part the part done by the compressing pump. The atmospheric line is *a b c*. The pump volume is *a b*, the motor volume is *a c*. The pump takes in the volume *a b* at atmospheric pressure; it compresses it into an intermediate receiver, the compression line (adiabatic) is *b f*, passing into receiver, line *f e*. From the receiver it enters the cylinder at the constant pressure of compression on the line *e f g*, supply of heat cut off at *g*. Then expansion (adiabatic) to the point *c* atmospheric pressure. The part *b f g c* is the part available for work, the part *b f e a* representing the work of the compressing pump, which is deducted from the total motor cylinder diagram *a e g c*.

The total volume of air passed through the pump is v_o, volume swept by motor piston, v. So far as the heat operations are concerned, the part of the diagram to volume v_c may be disregarded; it represents the pressing of the compressed charge into the reservoir after reaching the maximum pressure of compression (it is called v_c because it is volume of compression). The admission to the motor cylinder is

identical, so that work done in pump in that part equals work done upon the motor piston.

In addition to the letters used in type I,

v_c is volume of compression.
v_p volume at point g on diagram.
p_c is pressure of compression.
t_c is temperature of compression.

The temperature, volume, and pressure letters are figured below the diagram to make matters clear. Compression is carried on from volume v_o at atmospheric pressure and temperature to volume v_c at pressure p_c and temperature t_c, the curve being abiabatic.

After compression, heat is added without allowing the pressure to increase, but the piston moves out till the maximum temperature T is attained, and the supply of heat being completely cut off, adiabatic expansion follows till the atmospheric pressure is reached; the exhaust valve is then opened, and the hot gases discharged.

It is evident that as the pressure is constant, while heat is being given, the amount of heat given to the engine in all is

$$\text{H} = \text{K}_p\,(\text{T} - t_c),$$

and the heat discharged from it is also at constant pressure,

$$\text{H}^1 = \text{K}_p\,(\text{T}^1 - t).$$

The efficiency is therefore

$$\text{E} = \frac{\text{K}_p\,(\text{T} - t_c) - \text{K}_p\,(\text{T}^1 - t)}{\text{K}_p\,(\text{T} - t_c)}$$

$$= 1 - \frac{\text{T}^1 - t}{\text{T} - t_c} \tag{4}$$

The compression and expansion curves being adiabatic,

$$\text{Compression } p_c\,v_c^\gamma = p\,v_o^\gamma,$$
$$\text{Expansion } p_c\,v_p^\gamma = p_o v^\gamma,$$

$$\therefore \quad \frac{v_c^\gamma}{v_p^\gamma} = \frac{p\,v_o^\gamma}{p_o\,v^\gamma}, \text{ but } p_o = p,$$

$$\text{so that } \frac{v_c^\gamma}{v_p^\gamma} = \frac{v_o^\gamma}{v^\gamma} \tag{a}$$

$$\text{and } \frac{v_c}{v_p} = \frac{t_c}{\text{T}}, \text{ also } \frac{v_o}{v} = \frac{t}{\text{T}^1}.$$

Substituting in equation (a)

$$\frac{t_c}{\text{T}} = \frac{t}{\text{T}^1},$$

$$\text{and } \frac{\text{T}^1}{\text{T}} = \frac{t}{t_c}.$$

As the efficiency is

$$E = 1 - \frac{T^1 - t}{T - t_c},$$

$$\text{it may be either} = 1 - \frac{T^1}{T} \text{ or} = 1 - \frac{t}{t_c} \qquad (5)$$

That is, when expansion is carried to the same pressure as existed before compression, the efficiency depends upon the compression alone, t being the temperature before compression, and t_c the tem-

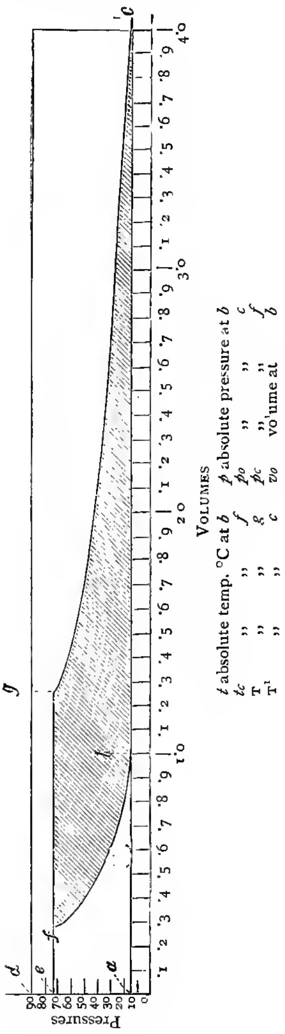


FIG. 24.—Type 2. Perfect diagram. Complete expansion

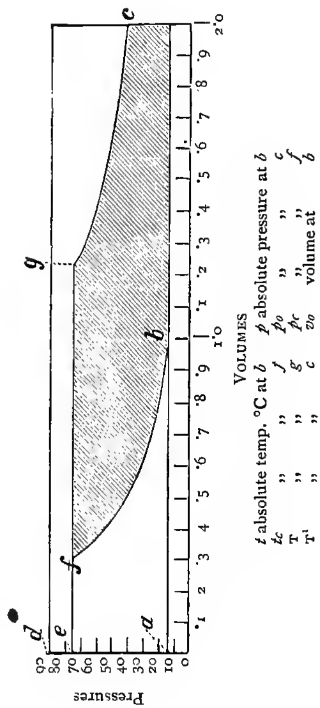


FIG. 25.—Type 2. Perfect diagram. Incomplete expansion

perature of compression. The efficiency being $1 - \frac{t}{t_c}$, the greater the temperature t_c the less is the fraction $\frac{t}{t_c}$, and the more nearly does E approach unity.

In most working engines of this kind, the expansion is not continued long enough to make the pressure after expanding fall to atmosphere; so that the efficiency is never so great, as when that is done a greater portion of the heat is discharged than need be. The modification of the formulæ is precisely as in type 1 for similar circumstances. A diagram of the kind is shown at fig. 25. The temperature t^1 is found as before:

$$t^1 = T^1 \frac{p}{p_o}.$$

The heat supplied to the cycle is as before:

$$H = K_p(T - t_c),$$

and the heat discharged is

$$H^1 = K_v\ (T^1 - t^1) + K_p\ (t^1 - t).$$

The efficiency is

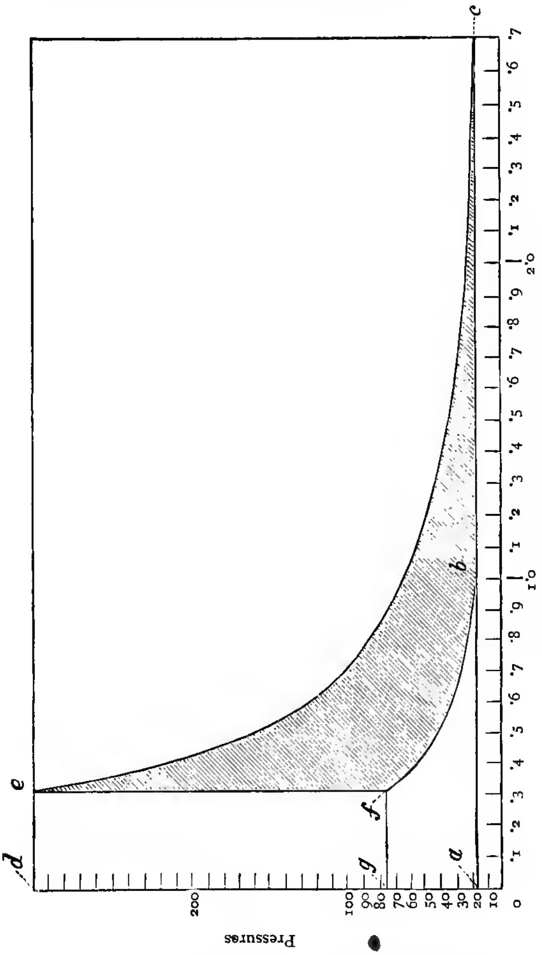
$$E = 1 - \frac{\frac{1}{\gamma}(T^1 - t^1) + (t^1 - t)}{T - t_c}. \tag{6}$$

Although there is no fixed proportion between the efficiency and the temperature of adiabatic compression, it is evident that E increases with increase of t_c.

Type 3.—A perfect indicator diagram of an engine of this type is shown at fig. 26. As in type 2, the diagrams of pump and motor are combined, the whole diagram being that given in the motor cylinder, but the shaded portion only represents the available work. The atmospheric line is *a b c*. The pump volume is *a b*, the motor cylinder volume is *a c*. The pump takes in the volume *a b* at atmospheric pressure, compresses it on the adiabatic line *b f* and into a receiver on the line *f g*. The compressed gases enter the motor cylinder on the line *g f*, heat is added instantaneously, and the pressure rises on the line *f e*. Supply of heat cut off at *e* and the expansion line *e c* is adiabatic. The total diagram in the motor cylinder is *a g f e c*, but the portion *a g f b* is common to motor and pump; the available work is therefore *b f e c*.

The total volume of air passed through the pump is v_o; the volume after adiabatic compression, from atmospheric pressure p and temperature t to pressure of compression p_c and temperature t_c, is v_c. Heat is supplied at constant volume v_c till the maximum temperature

of the explosion T is attained. The piston then expands the hot gases adiabatically from temperature T to T^1 and pressure P_o to pressure p_o, which in this case is equal to atmosphere.



t absolute temp. °C. at b p absolute pressure at b
t_c ,, ,, f p_o ,, ,, c
p_c ,, ,, f
P_o ,, ,, e
T ,, ,, e v_o volume at b
T^1 ,, ,, c v ,, ,, c
v_c ,, ,, f

FIG. 26.—Type 3. Perfect diagram. Complete expansion

The heat is discharged in passing from volume v to v_o at constant pressure of atmosphere. The part of the diagram from volume v_c to zero may be disregarded, as it is common to both pump and motor.

The heat supplied to the cycle is

$$H = K_v (T - t_c).$$

Heat discharged

$$H^1 = K_p (T^1 - t).$$

The efficiency is

$$E = \frac{K_v (T - t_c) - K_p (T^1 - t)}{K_v (T - t_c)}$$

$$= 1 - \gamma \frac{T^1 - t}{T - t_c}. \quad (7)$$

It is evident that for any maximum temperature T and compression temperature t_c there is a temperature T^1 at which the expansion adiabatic line falls to atmosphere. It will much simplify subsequent calculations to establish the relations between T, t_c, t and T^1.

$P_o v_c^\gamma = p_c v^\gamma$ and $p_c v_c^\gamma = p v_o^\gamma$ and as $p_o = p$,

$$\frac{P_o}{p_c} = \frac{v^\gamma}{v_o^\gamma}$$

but $$\frac{v}{v_o} = \frac{T^1}{t} \text{ so that } \frac{P_o}{p_c} = \frac{T^{1\gamma}}{t^\gamma}$$

and $$\frac{P_o}{p_c} = \frac{T}{t_c} \text{ so that } \frac{T}{t_c} = \left(\frac{T^1}{t}\right)^\gamma.$$

T^1 in terms of T, t_c and t is therefore

$$T^1 = t \left(\frac{T}{t_c}\right)^{\frac{1}{\gamma}} \quad (8)$$

Although this is the best case for the third type it is not the one commonly occurring in practice; no engine has as yet been arranged to expand the gases after explosion to the atmospheric pressure.

Fig. 27 is a perfect diagram of the most common case, namely, when the expansion is carried only so far that the heat is discharged when the volume is the same as that existing before compression. The formula of efficiency is exceedingly simple, and leads to a very apparent and nevertheless somewhat paradoxical result.

The heat supplied to the cycle is

$$H = K_v (T - t_c),$$

and the heat discharged is

$$H^1 = K_v (T^1 - t),$$

because the volume of the air is the same as that existing before compression, and therefore the heat necessary to bring the fluid back to its original state can be abstracted at constant volume.

The efficiency is

$$E = \frac{K_v (T - t_c) - K_v (T^1 - t)}{K_v (T - t_c)}.$$

$$= 1 - \frac{T^1 - t}{T - t_c} \quad (9)$$

As both curves are adiabatic, and pass through the same volume change,

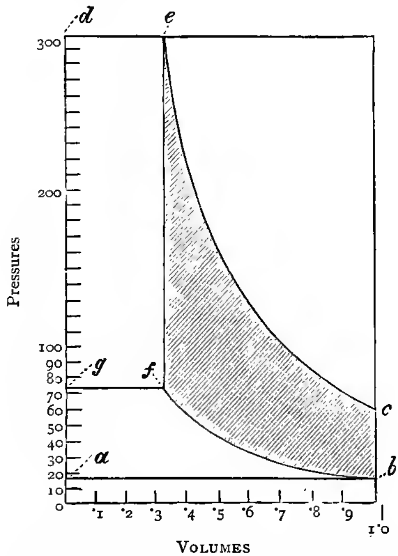
$$\frac{T^1}{T} = \frac{t}{t_c};$$

so that

$$\frac{T^1 - t}{T - t_c} = \frac{T^1}{T} = \frac{t}{t_c}.$$

The efficiency may therefore be expressed

$$E = 1 - \frac{T^1}{T} \text{ or } 1 - \frac{t}{t_c} \qquad (10)$$



t absolute temp. °C. at b — p absolute pressure at b
t_c „ „ f — p_c „ „ f
P_o „ „ e
T „ „ e — v_o volume at b
T^1 „ „ c — v „ c
v_c „ f
Here $v_o = v$

FIG. 27

Type 3. Perfect diagram. Expansion to same vol. as before compression

That is, the efficiency depends upon the ratio between the initial temperature and the temperature of adiabatic compression only. T, the temperature of explosion, may be any value greater than t_c, without either increasing or diminishing the efficiency. In this case

$$T^1 = \frac{Tt}{t_c}.$$

There is still another case of this type of cycle to be considered, when the expansion is continued beyond the original volume before

compression, but not carried far enough to reach atmospheric pressure. Fig. 28 is a diagram of the kind.

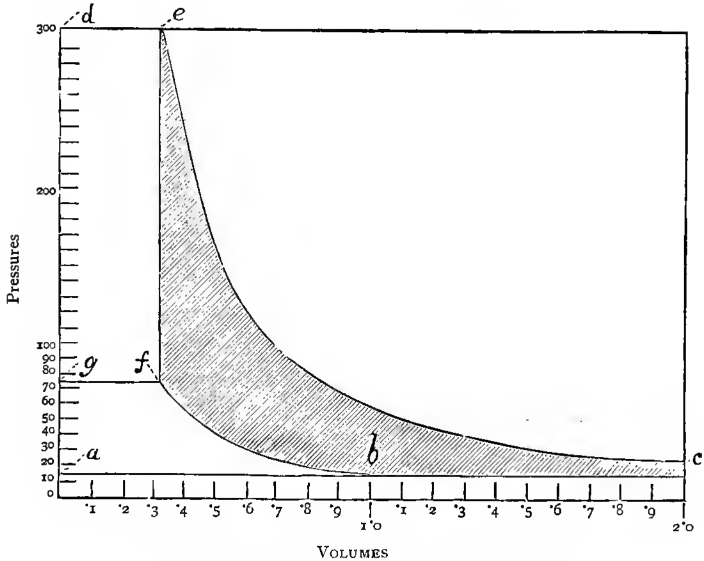
The heat supplied to the cycle is still

$$\text{H} = \text{K}_v\,(\text{T} - t_c).$$

The heat discharged may be found as in a similar case with types 1 and 2.

Total heat discharged is

$$\text{H}^1 = \text{K}_v\,(\text{T}^1 - t^1) + \text{K}_p\,(t^1 - t).$$



t	absolute temp. °C. at		b	p	absolute pressure at		b
t_c	,,	,,	f	p_c	,,	,,	f
				P_o	,,	,,	e
T	,,	,,	e	v_o	volume at		b
T[1]	,,	,,	c	v		,,	c
				v_c		,,	f

FIG. 28.—Type 3. Perfect diagram. Incomplete expansion

The efficiency is

$$\text{E} = \frac{\text{K}_v\,(\text{T} - t_c) - \{\text{K}_v\,(\text{T}^1 - t^1) + \text{K}_p\,(t^1 - t)\}}{\text{K}_v\,(\text{T} - t_c)}$$

$$= 1 - \frac{(\text{T}^1 - t^1) + \gamma\,(t^1 - t)}{\text{T} - t_c} \qquad (11)$$

Here then is no constant relationship between T^1 and T; the value of the cycle lies between cases 1st and 2nd. The efficiency is less than in the first case, but greater than in the second.

Type I A.—In this type of engine the efficiency cannot be stated in terms of temperature directly because of the nature of the perfect cycle.

The expansion line is adiabatic, and the compression line, whereby all the heat is discharged, is isothermal.[1]

Fig. 29 is the theoretical diagram of such an engine. The scale is altered from previous diagrams because of the great expansion.

There is no compression previous to the addition of heat; the heat is added at constant volume v_o, which is the volume of the charge. The pressure rises with the temperature from atmospheric pressure p and temperature t to maximum pressure P_o and temperature T. From T the expansion line is adiabatic, and is continued far enough to reduce the temperature again to t. The piston then returns, com-

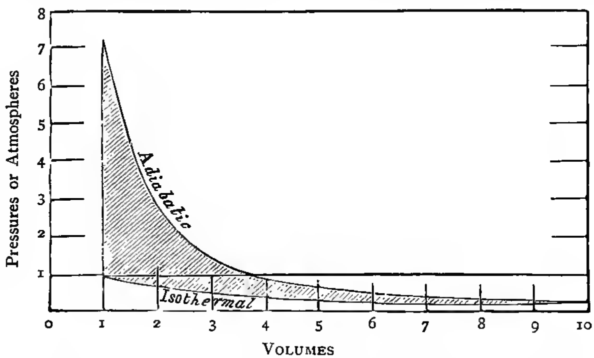


FIG. 29.—Type IA. Perfect diagram. Limited expansion

pressing the gases at the temperature t till the original volume v_o and pressure p are attained.

For any two temperatures t and T there is evidently a constant relationship between the available work and work discharged as heat. As in expanding from highest to lowest temperature the temperature falls from T to t, the whole area of the diagram T v_o v t may be taken as the heat supplied to the cycle.

The heat rejected is discharged at constant temperature t, and is equivalent to the area v_o v t t.

For any adiabatic curve the area T v_o v t is

$$\text{area} = \frac{1}{\gamma - 1}(P_o v_o - p_o v) \qquad (12)$$

[1] In Dr. A. Witz's able work, *Études sur les moteurs à gaz tonnant*, he falls into the error of supposing both expanding and compression lines of this type adiabatic, and he accordingly greatly over-estimates the efficiency proper to it.

For any isothermal

$$\text{area } v_o\, v\, t\, t = p\, v_o \text{ Log. }_\epsilon \frac{v}{v_o} \tag{13}$$

The efficiency is therefore—

$$E = \frac{\frac{1}{\gamma - 1}\left(P_o\, v_o - p_o\, v\right) - \left(p v_o \text{ Log. }_\epsilon \frac{v}{v_o}\right)}{\frac{1}{\gamma - 1}\left(P_o\, v_o - p_o v\right)}$$

$$= 1 - \frac{(\gamma - 1)\left(p v_o \text{ Log. }_\epsilon \frac{v}{v_o}\right)}{P_o\, v_o - p_o\, v} \tag{14}$$

but, as the line of compression discharging heat is an isothermal, that is, the temperature is kept constant at t during compression from the lowest pressure to atmosphere,

$$p v_o = p_o v \text{ (Boyle's law).}$$

The efficiency may therefore be written

$$E = 1 - \frac{(\gamma - 1)\left(p v_o \text{ Log. }_\epsilon \frac{v}{v_o}\right)}{P_o v_o - p v_o}$$

$$= 1 - \frac{(\gamma - 1)\, p \text{ Log. }_\epsilon \frac{v}{v_o}}{P_o - p}$$

then $$\frac{T}{t} = \frac{P_o}{p} = \left(\frac{v}{v_o}\right)^{\gamma - 1} \therefore \frac{v}{v_o} = \left(\frac{T}{t}\right)^{\frac{1}{\gamma - 1}}.$$

The efficiency can therefore be given entirely in terms of T and t:

$$E = 1 - \frac{(\gamma - 1)\, t \text{ Log. }_\epsilon \left(\frac{T}{t}\right)^{\frac{1}{\gamma - 1}}}{T - t} = 1 - \frac{t \text{ Log. }_\epsilon \frac{T}{t}}{T - t} \tag{15}$$

In the case where the expansion is not carried far enough to bring the temperature of explosion down to the temperature of the atmosphere, the efficiency can be found by using the formulæ 12 and 13 to get proportions of available and total work, and then get from the nature of the compression curve the total heat discharged. As this is variable, it will be better to study it from a numerical example later on.

The diagram given is the best possible for this kind of cycle.

Efficiency Formulæ for the Different Types

The general formulæ for efficiency of the four kinds of cycle are as follows:

TYPE 1, 1*st Case :*

$$E = 1 - \gamma \frac{T^1 - t}{T - t}. \tag{16}$$

T^1 in terms of T and t:

$$T^1 = t \left(\frac{T}{t}\right)^{\frac{1}{\gamma}}$$

2*nd Case :*

$$E = 1 - \frac{(T^1 - t^1) + \gamma\,(t^1 - t)}{T - t}. \tag{17}$$

TYPE 2, 1*st Case :*

$$E = 1 - \frac{T^1 - t}{T - t_c}; \tag{18}$$

$$\text{also } E = 1 \frac{t}{t_c}.$$

2*nd Case :*

$$E = 1 - \frac{\frac{1}{\gamma}(T^1 - t^1) + (t^1 - t)}{T - t_c}. \tag{19}$$

TYPE 3, 1*st Case :*

$$E = 1 - \gamma \frac{T^1 - t}{T - t_c} \tag{20}$$

T^1 in terms of T and t:

$$T^1 = t \left(\frac{T}{t_c}\right)^{\frac{1}{\gamma}}$$

2*nd Case :*

$$E = 1 - \frac{T^1 - t}{T - t_c}. \tag{21}$$

$$\text{also } E = 1 - \frac{t}{t_c}.$$

3*rd Case :*

$$E = 1 - \frac{(T^1 - t^1) + \gamma\,(t^1 - t)}{T - t_c}. \tag{22}$$

TYPE 1 A :

$$E = 1 - \frac{t\,(\gamma - 1)\,\text{Log.}_\epsilon \left(\frac{T}{t}\right)^{\frac{1}{\gamma - 1}}}{T - t} = 1 - \frac{t\,\text{Log.}_\epsilon \frac{T}{t}}{T - t}. \tag{23}$$

Those formulæ will be found very convenient in rapidly calculating the theoretical efficiency for any kind of diagram, but they do not throw much light upon the relative advantage of the different types. In type 1, for instance, it is apparent that efficiency increases with increase of temperature because the fraction $\frac{T^1 - t}{T - t}$ becomes

less with increase of T, but it does not rapidly become less because T^1 also increases with increase of T.

In type 2, 1st case, the efficiency is quite independent of T, and is dependent only on the ratio between t and t_c or v_o and v_c. Increase of T (maximum temperature) increases the available portion of the engine diagram, and therefore the average pressure, but without altering the efficiency.

TYPE 3.—With this type it is easy to see (1st case) that the efficiency is greater than in type 1, but only a numerical example will show the proportion.

In the second case it may be greater or less than in type 1, depending altogether on the amount of the compression.

Of all these working cycles, engines operating in accord with type 3, 2nd case, have proved of overwhelming technical importance, and those using type 2, 1st case, are of great scientific interest, while the 2nd case of type 2 is also of some practical importance. It is accordingly desirable to consider further those cycles which require compression of the charge as part of the working operation.

FURTHER DISCUSSION OF COMPRESSION CYCLES

For engines employing adiabatic compression and expansion, there are three symmetrical types of thermodynamic cycle which are each cycles of maximum thermal efficiency for the conditions assumed. They may be called :

Constant temperature type,
Constant pressure type,
and Constant volume type.

In the *constant temperature type*, adiabatic compression raises the temperature through the entire range from lower to upper ; the whole of the heat is received during isothermal expansion at the upper temperature, adiabatic expansion reduces the working fluid from the upper to the lower temperature ; and that portion of heat which is discharged is rejected by isothermal compression at the lower temperature.

This is the Carnot cycle.

Here the efficiency E is the maximum possible between the limits in accordance with the second law of thermodynamics, and it is

$$E = \frac{T - T^1}{T} = 1 - \frac{T^1}{T},$$

where T is absolute temperature at which heat is supplied and T^1 absolute temperature at which heat is discharged.

In this case t, the temperature before compression, is equal to T$'$, and t_c, the temperature after compression, is equal to T, so that

$$E = 1 - \frac{t}{t_c}.$$

The Carnot cycle is unsuitable for use in a practicable engine, as will be shown later, but it is interesting to note that its efficiency is determined by compression, as stated already with reference to cases of types 2 and 3.

In the *constant pressure type*, adiabatic compression raises the pressure from the lower to the higher limit; the heat-supply is added at the upper constant pressure, and therefore at increasing temperature; adiabatic expansion reduces the working fluid from the upper to the lower constant pressure; and that portion of heat which is discharged is rejected at the lower constant pressure and therefore at diminishing temperature.

As has already been shown, here also the thermal efficiency is determined by the amount of compression; it is also

$$E = 1 - \frac{t}{t_c}.$$

In this case the maximum temperature of the working fluid is higher than t_c, as well as the temperature at the end of adiabatic expansion, but nevertheless for equal compression ratios its thermal efficiency has the same absolute value as that of the constant temperature type.

In the *constant volume type* adiabatic compression raises the temperature of the working fluid through a certain range; the heat-supply is added above that range at constant volume, so that pressure increases during heat addition as well as temperature; adiabatic expansion reduces the temperature through a certain range and increases the volume to that existing before compression; and that portion of the heat which is discharged is rejected at constant volume (the maximum volume) and at diminishing temperature.

In this case also, as has been already shown, the thermal efficiency is determined by the amount of compression; it is also

$$E = 1 - \frac{t}{t_c}.$$

Where t is the temperature and v the volume before compression, and t_c the temperature and v_c the volume after adiabatic compression, it can be shown that $\left(\frac{v_c}{v}\right)^{\gamma-1} = \frac{t}{t_c}$, so that E may be written

$$E = 1 - \left(\frac{v_c}{v}\right)^{\gamma-1}$$

and if $\frac{v_c}{v} = \frac{1}{r}$ the compression ratio then

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma-1}$$

That is, in all three symmetrical cyles—*constant temperature, constant pressure*, and *constant volume*—the thermal efficiency depends only on the ratio of the maximum volume before compression to the volume after compression.[1] For the same compression ratio (adiabatic) of $\frac{1}{r}$ the efficiency is the same throughout. In the constant temperature or Carnot cycle the temperature range during adiabatic compression extends from the minimum to the maximum temperature, and so the full efficiency permitted by the second law is ideally possible; while in the constant pressure and constant volume cycles, the adiabatic temperature rise does not reach the maximum temperatures assumed, so that a greater theoretical efficiency is possible. The measure of efficiency, however, in the latter cases is still the range of adiabatic temperature rise.

From this it follows that any desired efficiency may be obtained from the cycles by selecting a suitable compression ratio.

This discussion suggests the interesting reflection that compression is essential to any thermodynamic cycle requiring high efficiency.

Again it is desirable to say that the three cycles described are all cycles of maximum efficiency for the conditions stated, and they are cycles of equal efficiency for equal ratios of adiabatic compression.

To obtain a clear idea of the relative values of the efficiencies, it is necessary to calculate a few numerical examples.

Calculated Examples of Efficiency of the Types

Numerical Examples.—Using air as the working fluid, the value of γ, the ratio of specific heat at constant volume to specific heat at constant pressure, is 1·408.

$$\frac{K_p}{K_v} = \gamma = 1{\cdot}408.$$

The gaseous mixture used in a gas engine differs considerably from pure air in its composition, and consequently in the ratio between specific heat at constant volume and specific heat at constant pressure, but it is advisable in the first place to consider the

[1] This interesting generalisation was first stated by Professor H. L. Callendar to the Institution of Civil Engineers Committee on the Thermal Standard for Internal Combustion Engines in 1904.

cycle as using air pure and simple. So many circumstances modify the theoretical efficiency in actual practice that they can be best considered after studying the simpler cases.

The temperature 1600° C. is a very usual one in the cylinder of a gas engine, and it will be calculated in each instance as the maximum, 17° C. being taken as atmospheric temperature.

A similar set with 1000° C. as the maximum will be calculated to show in each case the change of efficiency, if any, with change of maximum temperature.

TYPE I.—1*st Case.* The expansion is continued to atmospheric pressure.

Taking $T = 1600°$ C. $= 1873°$ absolute.

$t = 17°$ C. $= 290°$ "

Then $T^1 =$ the temperature after adiabatic expansion to atmospheric pressure.

$$T^1 = t\left(\frac{T}{t}\right)^{\frac{1}{\gamma}} \qquad (2)$$

$$T^1 = 290\left(\frac{1873}{290}\right)^{\frac{1}{1\cdot408}} = 1090° \text{ absolute.}$$

The efficiency is

$$E = 1 - \gamma\frac{T^1 - t}{T - t} = 1 - 1\cdot408\ \frac{1090 - 290}{1873 - 290} = 0\cdot29$$

$E = 0\cdot29$ with maximum temperature of 1600° C.

Taking the maximum temperature of explosion as 1000° C.

Absolute

$T = 1273° = 1000°$ C.

$t = 290° = 17°$ C.

then $T^1 = 829°$.

$$E = 1 - 1\cdot408\ \frac{829 - 290}{1273 - 290} = 0\cdot23.$$

$E = 0\cdot23$ with maximum temperature of explosion as 1000° C.

In this cycle the efficiency evidently increases with increase of the temperature of the explosion, but not in proportion to the increase of temperature; a change of maximum temperature from 1000° to 1600° C. only causing the efficiency to rise from 0·23 to 0·29. That is, at the first temperature, 23 heat units out of every 100 given to the cycle will be converted into work, while with the second much higher temperature only 29 units of 100 will be converted into work.

The second case of this type is the one most commonly occurring in practice. The cylinder is so arranged that the charge is taken in for half-stroke, the explosion then occurs, and the piston completes its stroke, expanding the heated gases from one volume to two volumes.

In the diagram, fig. 23, suppose volume v to be equal to $2\ v_o$, and

$$T = 1873^\circ \text{ absolute.}$$
$$t = 290^\circ \quad ,,$$

To get T',

$$\frac{T}{T^1} = \left(\frac{v}{v_o}\right)^{\gamma-1} \text{ or } \frac{T^1}{T} = \left(\frac{v_o}{v}\right)^{\gamma-1}$$

$$T^1 = T\left(\frac{v_o}{v}\right)^{\gamma-1}$$

$$T^1 = 1873\left(\frac{1}{2}\right)^{0.408} = 1411^\circ \text{ absolute.}$$

To calculate efficiency t^1 is still required; it is, in terms of volume and t,

$$t^1 = \frac{v}{v_o}t = \frac{2}{1}290 = 580^\circ \text{ absolute.}$$

The efficiency can now be obtained from formula (17).

$$E = 1 - \frac{(T^1 - t^1) + \gamma\,(t^1 - t)}{T - t}$$

$$= 1 - \frac{(1411 - 580) + 1.408\,(580 - 290)}{1873 - 290}$$

$$= 1 - \frac{831 + 1.408 \times 290}{1583} = 0.22 \text{ nearly.}$$

For this case $E = 0.22$,
showing the effect of limiting the expansion and discharging at a pressure above atmosphere.

Taking the same ratio of expansion and the lower maximum temperature of 1000° C.

$$T = 1273^\circ \text{ absolute.}$$
$$t = 290^\circ \quad ,,$$

as before, $T^1 = T\left(\frac{v_o}{v}\right)^{\gamma-1} = 1273\left(\frac{1}{2}\right)^{0.408} = 959^\circ$ absolute,

and t^1 is still $290 \times 2 = 580^\circ$ absolute.

Therefore $E = 0.20$.

Here the diminution of efficiency due to diminished expansion is not so great as in the case of the first, or rather the higher, temperature,

with complete expansion	1000° C. giving	0·23,
,, limited ,,	1000° C. ,,	0·20;
with the higher temperature of	1600° C.,	
with complete expansion	1600° C. giving	0·29,
,, limited ,,	1600° C. ,,	0·22.

It is evident from these results that where the amount of expansion is from one volume to two volumes, as in the Lenoir and Hugon engines, the efficiency does not substantially improve with increasing temperature.

TYPE 2.—1*st Case*. Where the expansion is carried far enough to reduce the working pressure to atmosphere, the efficiency of this kind of engine is quite independent of the temperature of combustion. This is shown by Professor Rankine[1] in his work on the steam engine. Whether the heat added after compression be great or small in amount, the proportion of it which is converted into work is stationary.

In this kind of engine assume a compression of 60 lbs. per sq. in. above atmosphere, 75 lbs. per sq. in. absolute, taking the atmospheric pressure as 15 lbs. per sq. in.

The compression is, as before stated, adiabatic; no heat is lost or gained. The temperature rises simply because of work performed upon the air.

Let

Atmospheric temperature and pressure (absolute) $t, p = 290°$ and 15 lbs.
Compression pressure (absolute) $p_c =$ 75 lbs.
Temperature of compression (absolute) $= t_c$

$$\frac{t_c}{t} = \left(\frac{p_c}{p}\right)^{\frac{\gamma - 1}{\gamma}}$$

$$t_c = t\left(\frac{p_c}{p}\right)^{\frac{\gamma - 1}{\gamma}}$$

$$t_c = 290\left(\frac{75}{15}\right)^{0{\cdot}29} = 462{\cdot}5° \text{ absolute,}$$

$$E = 1 - \frac{290}{462{\cdot}5} = 0{\cdot}37$$

$$E = 0{\cdot}37.$$

This result is much better than any obtained with the first type. It holds equally good for all combustion temperatures; with either 1000° C. or 1600° C. the efficiency would still be 0·37, so long as that degree of compression was used. With a higher compression the efficiency increases; 100 lbs. per sq. in. above atmosphere is quite a workable degree of compression. It is instructive to calculate the efficiency with this pressure:

$t = 290°$ absolute.
$p =$ 15 lbs. per sq. in. absolute.
$p_c = 115$ „ „ „ „

[1] *The Steam Engine*, Prof. Rankine, p. 373, Formula (7).

$$t_c = 290 \left(\frac{115}{15}\right)^{0\cdot29} = 524^\circ \text{ nearly.}$$

$$E = 1 - \frac{290}{524} = 0\cdot45.$$

$$E = 0\cdot45.$$

This type is evidently much superior to the first type, as it is capable of greatly increased efficiency by the mere increase of compression.

In the engines in practice before 1890 expansion had not been carried far enough to give the results calculated above. It had been usual to construct the engine so that the compression pump was one-half of the volume of the motor cylinder, that is, the ratio of the expansion was from one volume to two volumes at atmosphere. Taking first a compression of 60 lbs. per sq. in. above atmosphere with this proportion between the volumes at atmosphere, and the highest temperature as 1600° C., then (diagram, fig. 25)

$$\begin{aligned} T &= 1873^\circ \text{ absolute.} \\ t &= 290^\circ \quad ,, \\ t_c &= 462\cdot5^\circ \quad ,, \\ t^1 &= 290 \times 2 = 580. \end{aligned}$$

Before getting T^1 it is necessary to get the volume v_p at the highest temperature. It is

$$v_p = v_c \frac{T}{t_c}$$

and $$v_c = v_o \left(\frac{p}{p_c}\right)^{\frac{1}{\gamma}} = 1 \left(\frac{15}{75}\right)^{\frac{1}{1\cdot408}} = 0\cdot318$$

$$\therefore\ v_p = 0\cdot318 \ \frac{1873}{462\cdot5} = 1\cdot29$$

and $$T^1 = T \left(\frac{v_p}{v}\right)^{\gamma-1} = 1873 \left(\frac{1\cdot29}{2}\right)^{0\cdot408} = 1566^\circ \text{ absolute.}$$

The efficiency can now be found by formula (19)

$$E = 1 - \frac{\frac{1}{\gamma}(T^1 - t^1) + (t^1 - t)}{T - t_c} = 1 - \frac{\frac{1}{\gamma}(1566 - 580) + (580 - 290)}{1873 - 462\cdot5}$$

$$= 1 - \frac{0\cdot71\ (986) + 290}{1410\cdot5} = 1 - 0\cdot70 = 0\cdot30$$

$$E = 0\cdot30.$$

Here the insufficient expansion has caused the efficiency possible from the compression to fall from 0·37 to 0·30.

Calculating in the same way for the greater compression of 100 lbs. per sq. in. above atmosphere, with expansion ratio between compression and motor cylinders of two, it is found that the result is improved.

Here $v_c = 0{\cdot}235$ vol.

and $v_p = 0{\cdot}841$ vol.

$$T^1 = T\left(\frac{v_p}{v}\right)^{\gamma - 1} = 1873\left(\frac{0{\cdot}841}{2}\right)^{0{\cdot}408} = 1318^\circ \text{ absolute.}$$

$$T = 1873^\circ$$
$$t = 290^\circ$$
$$t^1 = 580^\circ$$
$$t_c = 524^\circ$$

The efficiency is therefore

$$E = 1 - \frac{\frac{1}{\gamma}(T^1 - t^1) + (t^1 - t)}{T - t_c} = 1 - \frac{0{\cdot}71\ (1318 - 580) + (580 - 290)}{1873 - 524}$$

$$= 1 - \frac{0{\cdot}71 \times 738 + 290}{1349} = 1 - \frac{814}{1349} = 0{\cdot}40$$

$$E = 0{\cdot}40.$$

The great compression has greatly increased the efficiency while leaving the proportion of the two cylinders unaltered.

Still using the same cylinders, the efficiency with compression of 60 lbs. above atmosphere and a maximum temperature of 1000° C., is

$$E = 0{\cdot}36 \text{ nearly,}$$

the data being

$$T^1 = 906^\circ \qquad T = 1273^\circ$$
$$t^1 = 580^\circ \qquad t = 290^\circ$$
$$t_c = 462^\circ,$$

volumes

$$v_o = 1 \qquad v = 2$$
$$v_c = 0{\cdot}318 \qquad v_p = 0{\cdot}87.$$

Using the higher compression 100 lbs. above atmosphere with 1000° C. as highest temperature

$$E = 0{\cdot}44.$$

Data: $T^1 = 763^\circ$ $\quad T = 1273^\circ$
$t^1 = 580^\circ$ $\quad t = 290^\circ$
$t_c = 524^\circ$

Vol.: $v_o = 1$ $\quad v = 2$
$v_c = 0{\cdot}235$ $\quad v_p = 0{\cdot}57$

In this kind of engine the best result is always obtained when the expansion is carried to atmospheric pressure. The necessary proportion between the two cylinders, to accomplish this, depends on two things : the temperature of compression, and the temperature of combustion. The ratio between the cylinders should be

$$r = \frac{T}{t_c}.$$

With a temperature of compression of 462°, for instance, and a maximum of 1873° absolute $\left(\frac{1873}{462} = 4{\cdot}05\right)$ the volume of the motor cylinder would require to be 4·05 times that of the pump. With the increased compression giving 524° absolute $\left(\frac{1873}{524} = 3{\cdot}57\right)$ ratio of motor to pump 3·57 to 1.

With the lower maximum temperature of 1273° the ratios for the two compression values are

$$\frac{1273}{462} = 2{\cdot}75 \qquad \frac{1273}{524} = 2{\cdot}43 \text{ nearly.}$$

These figures explain why the efficiency varies so much with two cylinders of ratio 1 to 2 with change of maximum temperature and compression.

TYPE 3.—*1st Case.* In this case expansion is carried to atmosphere. It is evident from the formulæ that efficiency varies to some extent with maximum temperature of the explosion.

Taking first a maximum temperature of 1600° C., as in the last type calculated, with a pressure of compression 60 lbs. above atmosphere.

The data are as follows :

Temperatures $\quad T = 1873° \qquad t = 290$
$\qquad t_c = 462.$

T^1 in terms of T and t, t_c is (see p. 80)

$$T^1 = t\left(\frac{T}{t_c}\right)^{\frac{1}{\gamma}} = 290 \left(\frac{1873}{462}\right)^{\frac{1}{1{\cdot}408}} = 783°$$

$$T^1 = 783°.$$

The efficiency therefore

$$E = 1 - \gamma\frac{T^1 - t}{T - t_c} = 1 - 1{\cdot}408\,\frac{783 - 290}{1873 - 462}$$

$$E = 0{\cdot}51.$$

With compression 100 lbs. above atmosphere,

$$t_c = 524°$$

and T^1 is therefore $$T^1 = 290\left(\frac{1873}{524}\right)^{\frac{1}{1\cdot408}} = 716^\circ$$

and $$E = 1 - 1\cdot408\,\frac{545 - 290}{1873 - 524}$$

$$E = 0\cdot55.$$

Taking, next, 1000° C. as the highest temperature, first with the lower compression, and after with the higher compression,

with 60 lbs. compression T^1 is 595° absolute
with 100 ,, T^1 is 545° ,,
$E = 0\cdot47$ at 60 lbs., $E = 0\cdot52$ at 100 lbs., with 1000° C.

In this case the efficiency varies both with the maximum temperature of the explosion and with the compression temperature previous to explosion. A glance at the numbers placed together will show clearly the relationship.

Max. temps. in °C.	1600°	1600°	1000°	1000°
Pressure of compression above atmosphere	60 lbs.	100 lbs.	60 lbs.	100 lbs.
Efficiency	0·51	0·55	0·47	0·52

2nd Case.—Here the expansion after explosion is not carried on far enough to reduce the pressure to atmosphere. It terminates when the volume is the same as existed before compression, that is, the volume swept by the motor piston when the air is expanding doing work is identical with that swept by the pump piston in compressing up to maximum pressure. Pump and motor are equal in volume. To this case of type 3 belong all compression engines in which the motor piston compresses its charge into a space at the end of the cylinder. In this case, as in case 1, type 2, the theoretical efficiency of the engine is quite independent of the maximum temperature of the explosion. So long as the volume after expansion is the same as that before compression, it does not matter in the least how much heat is added at constant volume of compression; whether only a few degrees' rise occurs or 1000° or 2000°, it is all the same so far as the proportion of added heat converted into work is concerned. That proportion depends solely upon the amount of compression.

For 60 lbs. adiabatic compression, temperature 462° absolute, the efficiency is 0·37; for 100 lbs. above atmosphere it is 0·45. Given by the formula $$E = 1 - \frac{t}{t_c}.$$ (See p. 80)

E depends absolutely upon the temperature of the atmosphere and the temperature of compression t and t_c. If the relative volumes of space swept by piston and compression space be known, then the efficiency can be at once calculated.

3rd Case.—Here the expansion is carried further than the original volume before compression, but not far enough to reduce the pressure to atmosphere. Efficiency is always less than in the first case with corresponding temperature of explosion and compression, but greater than in the second case. It is found by the formula :

$$E = 1 - \frac{(T^1 - t^1) + \gamma (t^1 - t)}{T - t_c}.$$

t^1 depends on the relationship between the volumes v_o and v the volume at atmosphere and the volume of discharge after expansion. It is always :

$$t^1 = t\frac{v}{v_o}.$$

T^1 is also found by the same method as in types 1 and 2. It is better to postpone calculating any particular case of this at present, as no engine doing this has yet got into public use, and it can be considered further on in discussing the effect of increased expansion in the actual engines.

Type 1 A.—The efficiency of this type of heat cycle depends to a considerable extent upon cooling during the return stroke ; in its best form, cooling at the lowest temperature during isothermal compression, it cannot be carried out without introducing the very disadvantages with which the hot-air engine was saddled—namely, a dependence upon the slow convection of air for the discharge of the heat necessarily rejected from the cycle. The rapid performance of this operation is impossible, and accordingly it is hardly fair to compare this type with those preceding ; they could all of them be greatly improved in theory by introducing greater expansions and cooling by convection at the lowest temperature, but all at the expense of rate of working. The efficiency of type 1 A will be found to be high ; but it is to be kept constantly in mind that the penalty of slow rate of work was fully exacted in the practical examples of the kind in public use. They are exceedingly cumbrous, and give but a trifling power in comparison with their bulk and weight. The efficiency in this type is dependent upon T and t only.

$$E = 1 - \frac{(\gamma - 1)\, t \operatorname{Log.}_\epsilon \left(\frac{T}{t}\right)^{\frac{1}{\gamma - 1}}}{T - t} = 1 - \frac{t \operatorname{Log.}_\epsilon \frac{T}{t}}{T - t}.$$

Take first

$$T = 1873^\circ$$

$$t = 290^\circ$$

$$E = 1 - \frac{290 \times 1{\cdot}865}{1583}$$

$$E = 0{\cdot}66.$$

This is a very high efficiency, but it is obtained by using an enormous expansion,

$$\frac{v}{v_o} = \left(\frac{T}{t}\right)^{\frac{1}{\gamma - 1}} = 96 \cdot 7 \text{ nearly.}$$

The piston must move through nearly 100 times the original volume of the charge before the temperature is reduced to the temperature existing before igniting ; in passing back to unit volume the gases must be supposed to keep at t by the cooling effect of the cylinder walls.

When $T = 1000°$ C. $= 1273°$ absolute,
$t = 17°$ C. $= 290°$,,

the efficiency is $E = 0{\cdot}56$,

and the expansion required is not so great, being 37·5 volumes.

The actual ratios of expansion used in practice have not approached those proportions, and will be considered while discussing the diagrams taken from engines of this type.

Further Calculations on Compression Cycles

It has been shown that the thermal efficiency of each of the three symmetrical thermodynamic cycles—*constant temperature, constant pressure*, and *constant volume*—can be calculated from the adiabatic compression ratio $\frac{1}{r}$ by the formula

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma - 1}$$

With air as the working fluid $\gamma - 1 = 0{\cdot}408$,

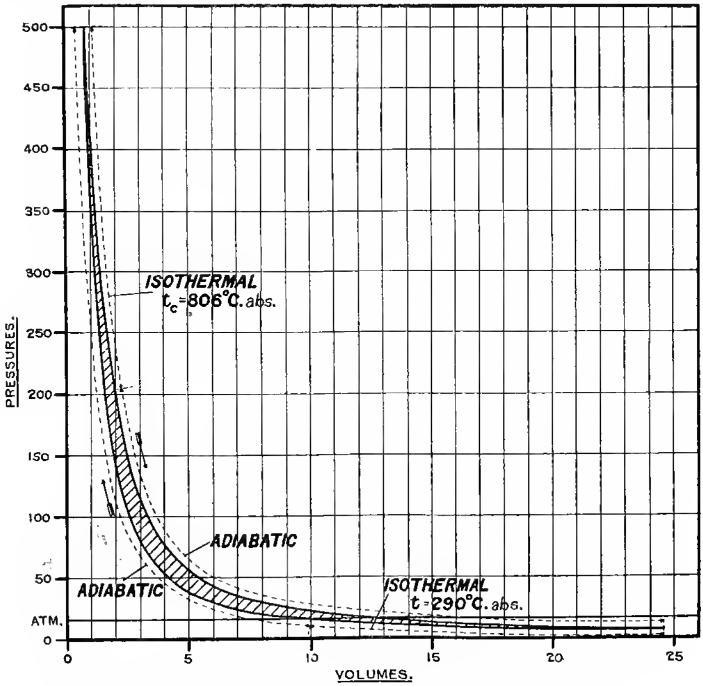
$$\text{and } E = 1 - \left(\frac{1}{r}\right)^{0 \cdot 408}$$

This formula has been employed to calculate various values of E when $\frac{1}{r}$ varies from $\frac{1}{2}$ to $\frac{1}{100}$. This gives a table of efficiencies for the three cycles as follows :

Table of Theoretic Thermal Efficiency for the three Symmetrical Cycles of Constant Temperature, Pressure, and Volume.

$\frac{1}{r}$	E
$\frac{1}{2}$	0·246
$\frac{1}{3}$	0·36
$\frac{1}{4}$	0·43
$\frac{1}{5}$	0·48
$\frac{1}{7}$	0·55
$\frac{1}{10}$	0·61
$\frac{1}{20}$	0·70
$\frac{1}{100}$	0·85

From this it is obvious that any desired thermal efficiency may be theoretically obtained by increasing the compression. It must not be forgotten, however, that as compression increases so do compression temperatures, so that with a compression ratio of $\frac{1}{100}$ for example, the adiabatic compression temperature would rise from about 17° C. to 1600° C., so that little power could be attained for given engine dimensions.



Maximum pressure = 500 lbs. per sq. in. abs.; mean pressure = 6 lbs. per sq. in., efficiency = 0·64; vol. swept by piston = 2·5 times initial volume

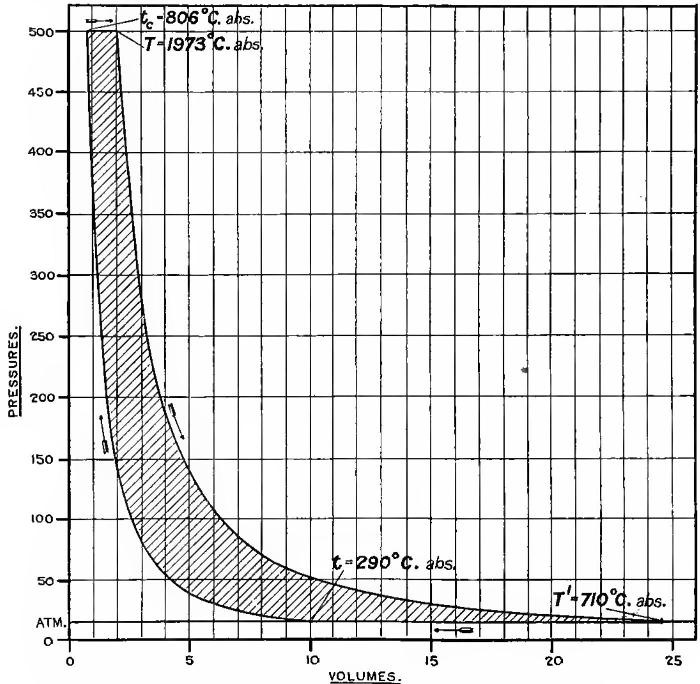
FIG. 30.—Diagram of Air Engine. Constant Temperature Type (Carnot Cycle)

The highest compression ratio used in practice is about $\frac{1}{12}$ in the Diesel engine. In ordinary practice the value lies between $\frac{1}{5}$ and $\frac{1}{7}$. Ideal diagrams have been calculated for the three cycles under such conditions that the maximum pressure of the working fluid is the same. This is the condition which would guide the engine designer, as the maximum pressure broadly determines the strength necessary for the engine. Figs. 30, 31, 32, and 33 show such diagrams.

In all these cases the temperature before adiabatic compression t is taken as 17° C.—that is, 290° C. absolute; the maximum pressure

chosen for comparison is 500 lbs. per sq. in. absolute, and pressure before compression atmospheric 14·7 lbs. per sq. in.

Taking first the *constant temperature type diagram*, fig. 30, there are four operations: (*a*) adiabatic compression; (*b*) isothermal expansion; (*c*) adiabatic expansion; and (*d*) isothermal compression



Maximum pressure = 500 lbs. per sq. in. abs.; mean pressure = 56 lbs. per sq. in. efficiency = 0·64; vol. swept by piston = 2·5 times initial volume.

FIG. 31.—Diagram of Air Engine. Constant Pressure Type

to initial volume. The four operations are marked on the diagram, and the direction of motion of the piston is indicated by arrows.

Here $t = 290°$ C. absolute.

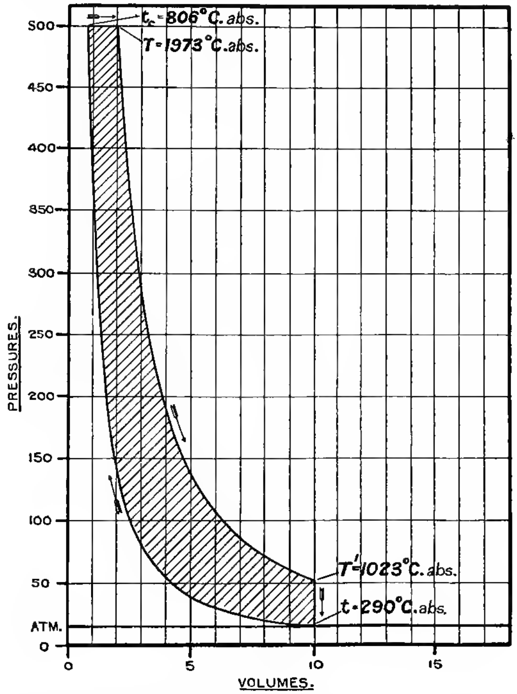
and $t_c = 806°$ C. ,,

$$\frac{1}{r} = \frac{1}{12 \cdot 24}$$

$$E = 1 - \frac{290}{806} \text{ or } 1 - \left(\frac{1}{12 \cdot 24}\right)^{0 \cdot 408} = 0 \cdot 64.$$

Here P_o and p_c are equal, 500 lbs. per sq. in. absolute. Maximum volume of cycle is 2·5 times initial volume and the mean effective

pressure is 6 lbs. per sq. in. This result at once shows why the Carnot cycle is impracticable : only 6 lbs. effective pressure per sq. in. throughout the working stroke is obtained in an ideal engine in which the necessary maximum pressure is 500 lbs. per sq. in. absolute. The acting maximum pressure for which strength of parts must be provided is $500 - 15 = 485$, and the ratio of maximum to mean effective is $\frac{485}{6} = \frac{80 \cdot 8}{1}$, say 81 to 1. The strength of engine parts to resist static



Maximum pressure = 500 lbs. per sq. in. abs. ; mean pressure = 117 lbs. per sq. in. ; pressure at end of expansion = 51·8 lbs. per sq. in. abs. ; efficiency = 0·56.

FIG. 32.—Diagram of Air Engine. Constant Pressure Type (Second Case)

pressures only is thus 81, while the effective pressure for power is represented by 1. A very large and heavy engine would be required to produce a very small power.

Taking, second, the *constant pressure type diagram*, fig. 31, there are four operations : (*a*) adiabatic compression ; (*b*) expansion at constant upper pressure and increasing temperature ; (*c*) adiabatic expansion to lower pressure ; (*d*) compression at lower pressure temperature falling to original temperature restoring volume to initial volume.

The operations and direction of piston motion are indicated as before.

Here $t = 290°$ C. absolute.

and $t_c = 806°$ C. ,,

$T = 1{,}973°$ C. ,,

$T^1 = 710°$ C. ,,

$\frac{1}{r} = \frac{1}{12{\cdot}24}$

and $E = 0{\cdot}64$ as in the previous case.

Here P_o and p_c are again equal and the pressure is 500 lbs. per sq. in. absolute. But the mean effective pressure is much greater, it is 56 lbs. per sq. in., so that with the same acting maximum pressure of 485 lbs. the ratio of maximum to mean effective is very much more favourable, it is

$$\frac{485}{56} = 8{\cdot}66$$

Here obviously we can get very much more power for a given weight of metal.

Fig. 32 shows a modification of this type giving a much higher power for a given weight with but a small change of efficiency; it represents the second case of Type 2, using the maximum pressure of 500 lbs. per sq. in., and terminating the adiabatic expansion when the volume attains the initial volume before compression. Here the heat discharged from the cycle is rejected at constant volume, and the efficiency formula becomes more complicated. The pressure is constant while heat is being received by the working fluid, so that the heat added is

$$H = K_p (T - t_c).$$

and the heat discharged is at constant volume

$$H^1 = K_v (T^1 - t)$$

The efficiency is therefore

$$E = 1 - \frac{K_v (T^1 - t)}{K_p (T - t_c)}$$

or

$$E = 1 - \frac{T^1 - t}{1{\cdot}408 (T - t_c)}$$

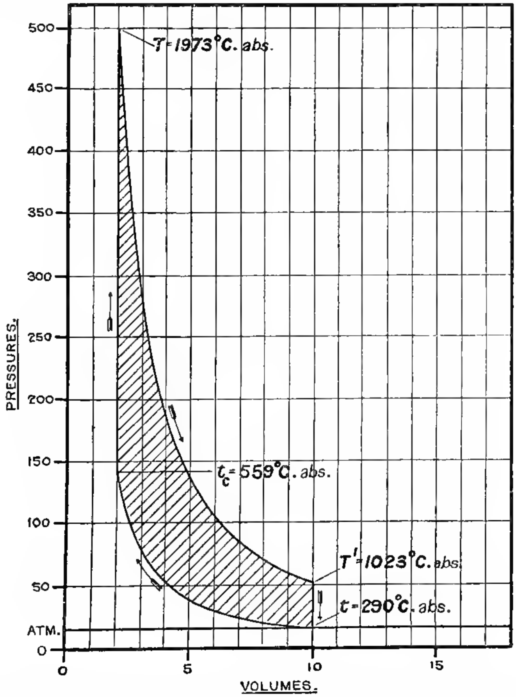
In this case

$$E = 1 - \frac{1013 - 290}{1{\cdot}408 (1973 - 806)} = 0{\cdot}56$$

The limitation of the expansion reduces the efficiency from 0·64 to 0·56, but it gives much more power for equal weight, as the mean effective pressure is 117 lbs. per sq. in.

The ratio of maximum to mean effective is $\frac{485}{117} = 4{\cdot}14$.

Taking, third, the *constant volume diagram*, fig. 33, there are four operations : (*a*) adiabatic compression ; (*b*) addition of heat at constant volume and rising temperature ; (*c*) adiabatic expansion ; (*d*) rejection of heat at constant volume falling to initial temperature.



Maximum pressure = 500 lbs. per sq. in. abs. ; mean pressure = 105 lbs. per sq. in. ; compression pressure = 141·8 lbs. per sq. in. abs. ; pressure at end of expansion = 51·8 lbs. per sq. in. abs. ; efficiency = 0·48.

FIG. 33.—Diagram of Air Engine. Constant Volume Type

The operations and the direction of piston movement are indicated as before

Here

$$t = 290° \text{ C. absolute.}$$
$$t_o = 559° \text{ C.} \quad ,,$$
$$T = 1973° \text{ C.} \quad ,,$$
$$T^1 = 1013° \text{ C.} \quad ,,$$
$$\frac{1}{r} = \frac{1}{5}$$

$$E = 1 - \frac{290}{559} \text{ or } 1 - \left(\frac{1}{5}\right)^{0{\cdot}408} = 0{\cdot}48.$$

Then P_o is 500 lbs. per sq. in. absolute, and p_c is 141·8 lbs. per sq. in. absolute. Maximum volume of cycle is the initial volume before compression.

The mean effective pressure is 105 lbs. per sq. in., and ratio of maximum to mean effective $\frac{485}{105} = 4·6$.

Constant pressure and constant volume types at the same compression. Fig. 34 is a diagram combining constant pressure and constant volume types employing the same compression and the same maximum temperature.

In both cases—

$$t = 290° \text{ C. absolute}$$
$$t_c = 559° \text{ C.} \quad ,,$$
$$T = 1973° \text{ C.} \quad ,,$$
$$T^1 = 1023° \text{ C.} \quad ,,$$
$$\frac{1}{r} = \frac{1}{5}$$

and

$$E = 0·48$$

In the *constant pressure* diagram the maximum pressure is 141·8 lbs. absolute per sq. in., the mean effective pressure throughout the stroke is 37 lbs. per sq. in. The effective maximum is $141·8 - 15 = 126·8$ lbs. Ratio of maximum to mean effective $\frac{126·8}{37} = 3·8$. The maximum volume of the cycle is 3·5 times the initial volume at 290° C. absolute.

In the *constant volume* diagram it has been already pointed out that the effective maximum is 485 lbs. per sq. in., the mean effective 105 lbs., and the ratio of maximum to mean 4·6, while the maximum volume is equal to the initial volume before compression. Here we have two diagrams of equal thermal efficiency, of which the *constant pressure* type requires an engine of 3·5 times the cylinder capacity of the *constant volume*, but where the ratio of maximum to mean is more favourable, 3·8 against 4·6, so that for equal power the *constant pressure* engine would require to withstand smaller stresses than the *constant volume.* Practical considerations alone would determine here which cycle should be adopted. Such calculations cannot be made until we study the real as well as the ideal conditions assumed.

Comparison of Results

The three maximum temperatures used, 1700° C., 1600° C., and 1000° C., with the lowest temperature, 17° C., give, in a perfect heat engine, efficiencies

$$1700° \text{ C.} = 0·853$$
$$1600° \text{ C.} = 0·85 \text{ nearly,}$$
$$1000° \text{ C.} = 0·77 \quad ,,$$

To compare easily the following table will be useful:

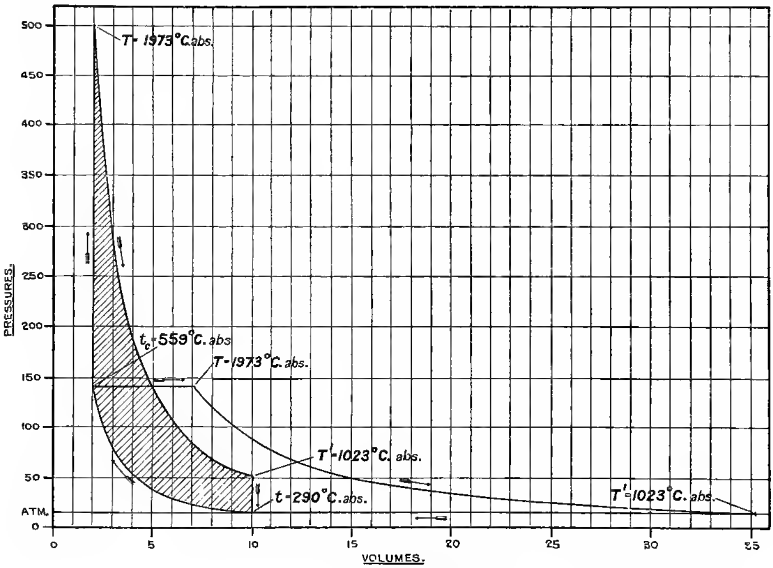
TABLE OF THEORETIC EFFICIENCIES CALCULATED IN THE EXAMPLES GIVEN

—	Max. temp. °C abs	Compression Temp °C. abs.	Compression Ratio $\frac{1}{r}$	E
Type 1				
Expanding to atmosphere	1873°	—	—	0·29
,, ,, ,,	1273°	—	—	0·23
Expanding to twice initial volume	1873°	—	—	0·22
	1273°	—	—	0·20
Type 2				
Expanding to atmosphere	—	462°	$\frac{1}{3\cdot1}$	0·37
	—	524°	$\frac{1}{4\cdot4}$	0·45
Expanding to twice volume existing before compression	1873°	462°	—	0·30
	1873°	524°	—	0·40
	1273°	462°	—	0·36
	1273°	524°	—	0·44
Type 3				
Expanding to atmosphere	1873°	462°	—	0·51
	1873°	524°	—	0·55
	1273°	462°	—	0·47
	1273°	524°	—	0·52
Expanding to the same volume as existed before compression	—	462°	$\frac{1}{3\cdot1}$	0·37
		524°	$\frac{1}{4\cdot4}$	0·45
Type 1A				
Expanding from maximum temperature to lowest temperature	1873°	—	—	0·66
	1273°	—	—	0·56

Symmetrical cycles of *constant temperature, constant pressure,* and *constant volume.*

$\frac{1}{r}$	E
$\frac{1}{2}$	0·246
$\frac{1}{3}$	0·36
$\frac{1}{4}$	0·43
$\frac{1}{5}$	0·48
$\frac{1}{7}$	0·55
$\frac{1}{10}$	0·61
$\frac{1}{20}$	0·70
$\frac{1}{100}$	0·85

Comparing these types it is evident that types 2 and 3, which employ compression previous to ignition or heating, are far superior to types 1 and 1 A, which employ no previous compression. It is not true, however, that the two latter are both non-compression, because 1 A involves not only great expansions, large volumes at low pressures, but also recompression after expansion, this recompression being effected isothermally at the lower temperature limit. This is an impracticable process for long expansions and consequently



	Constant pressure.	Constant volume.
Compression pressure	141·8 lbs. per sq. in. abs.	141·8 lbs. per sq. in. abs.
Maximum pressure	141·8 lbs. per sq. in. abs.	500 lbs. per sq. in. abs.
Mean pressure	37 lbs. per sq. in.	105 lbs. per sq. in.
Pressure at end of expansion .	14·7 lbs. per sq. in. abs.	51·8 lbs. per sq. in. abs.
Efficiency	0·48	0·48

FIG. 34.—Diagrams of Air Engines

Constant Pressure and Constant Volume Types, with the same compression and the same maximum temperature.

correspondingly long compressions. High theoretical efficiencies, however, are attainable by the method. Type 1 has neither prior nor subsequent compression, and so gives only low thermal efficiencies, because only small expansions are possible. These two types may be dismissed from further consideration.

The three symmetrical cycles of *constant temperature*, *constant pressure*, and *constant volume* are all perfect cycles for their assumed conditions and all three are theoretically capable of yielding thermal

efficiencies which depend solely on the degree of adiabatic compression employed. Of the three, however, the first is impracticable because of the very low effective mean pressure obtainable compared with the very high relative maximum pressure. In the example calculated, the ratio is 81 to 1, a quite impossible ratio for a practicable engine. It must always be borne in mind in reasoning on these thermodynamic cycles that the compression ratio which may be chosen is dependent on the maximum temperature available. Thus a compression ratio of $\frac{1}{100}$ could not be adopted unless that maximum temperature were at least 1600° C., assuming the lower temperature as 17° C., because adiabatic compression from 100 to 1 volume would raise the temperature from 17° C. to 1600° C. In such a case the *constant temperature* cycle would be the only one available, as compression has already attained maximum temperature and heat at that temperature can only be added isothermally.

This consideration excludes such an extreme compression range as $\frac{1}{100}$, that is it confines practicable engines to types 2 and 3—that is, to *constant pressure* and *constant volume* cycles.

In practice the ratio now adopted varies between $\frac{1}{4}$ and $\frac{1}{13}$ in both cases, so that their thermal efficiencies vary between 0·43 and 0·65.

Quite favourable maximum and mean effective pressure ratios are obtained between these limits. In the example calculated for *constant pressure* conditions, as in fig. 32, it is 4·14 to 1, with an efficiency of 0·56. The corresponding *constant volume* diagram, fig. 33, gives 4·6 to 1 with an efficiency of 0·48; here the constant pressure cycle is best from both points of view. In both cases the total volumes of working fluid used are the same, and the maximum volume is the same. Comparing the cases shown on fig. 34 of *constant pressure* and *constant volume*, efficiencies both 0·48, the maximum to mean effective pressure ratio is 3·8 for *constant pressure* and 4.6 for *constant volume*, using identical volumes of working fluid, but the maximum volume required for the constant pressure engine is 3·5 times that of the constant volume. It is entirely a question for practice whether it is better to have a smaller engine with high pressures or a larger engine with low pressures but parts strained relatively less.

So far practice favours the smaller high-pressure engine, and with engines of small power the choice is justifiable. Whether this will be so in the case of large gas engines has not been yet settled.

The Otto cycle engines made under type 3 far outnumber all others in present use, but high economies are obtained by the Diesel engine operating under type 2, although on the Otto mechanical cycle also. Undoubtedly these two types, *constant volume* and *constant pressure*, are of vital importance to-day.

Throughout the present chapter the working fluid has been

assumed to be dry air obeying perfectly the laws of Charles and Boyle; its specific heat has also been assumed to be constant throughout the temperature range. Its specific heat at constant volume K_v has been taken as 0·168, and specific heat at constant pressure $K_p = 0{\cdot}237$ and

$$\frac{K_p}{K_v} = \frac{0{\cdot}237}{0{\cdot}168} = 1{\cdot}408 = \gamma.$$

It is now known that the specific heat of air is not quite constant between 0° and 1400° C. Holborn and Austin's recent determinations of specific heat at constant pressure by a calorimetric method and electrical heating show a rise, which, however, is not large. The mean K_p between 100° C. and 1400° C. is about 8 per cent. higher than that between 100° and 200° C.

Also the working fluid of the gas engine, as has already been pointed out, is not air but a mixture of nitrogen, carbonic acid, steam, and oxygen, so that, apart from chemical questions of combustion, the physical properties of the real working fluid are different from the idealised air which has been assumed in the foregoing formulæ and examples. Nevertheless the author considers it desirable to treat the problem of ideal efficiency in its most simple form, as many valuable lessons have been learned from the simplified conditions and many valuable deductions have been made in the past from calculations on the 'air standard' first introduced by the author in 1882.

But it must be remembered that the efficiencies and mean pressures determined by these calculations for ideal air are not the efficiencies and mean pressures which would be proper to the actual working fluid. The air standard supplies a relative but not an absolute standard, as will be discussed later on when the actual working fluid has been studied.

Meantime, however, it may be taken that the reasoning and conclusions reached in this chapter are valuable when properly used.

Conclusions.—The best cycles for maximum efficiency and maximum power for given stresses are those of *constant pressure* and *constant volume* in which adequate compression forms an essential feature. By such compression we are enabled to subject our heated working fluid to any predetermined effective expansion. Without adiabatic compression, effective expansion is restricted within very narrow limits; with compression our limits may be chosen with a view to a pre-arranged thermal efficiency.

Compression previous to ignition does for the gas engine what the condenser does for the steam engine. It extends the range of effective expansion. It does more than that, but we must first discuss the laws of heat loss before dealing with effects other than purely thermodynamic.

CHAPTER IV

THE CAUSES OF LOSS IN GAS ENGINES

In calculating the efficiency of the different kinds of engines, it has been assumed that the conditions peculiar to each cycle have been perfectly complied with. In actual engines this is impossible; it is therefore necessary to discover in what manner practice fails in performing the operations required by theory.

The actual engines differ from the ideal ones in several ways:

1. The working fluid loses heat to the walls enclosing it after its temperature has been raised to the highest point;

2. The working fluid often gains heat when entering the cylinder at a time when it should remain at the lowest temperature;

3. The supply of heat is never added instantaneously as is required in some types;

4. The working fluid is not the pure dry air assumed in Chapter III.; it is a mixture of nitrogen, carbonic acid, steam, and oxygen, with a different specific heat from air, which specific heat increases considerably with rise in temperature;

5. Combustion is not complete when the maximum temperature of the working fluid is attained, and in some cases it is not complete when the exhaust valve opens;

6. Combustion changes the volume of the working fluid so that the volume which is heated is different from the volume which is compressed;

7. The admission, transfer, and expulsion of the working fluid are not accomplished without some resistance, throttling during admission, back-pressure during exhaust;

8. Loss of heat to the cylinder walls during compression.

The first cause of loss is by far the most considerable and will be considered first.

Loss of Heat to the Cylinder and Piston

Although this is the most considerable source of loss in all gas engines, the stock of information in existence upon the subject is quite

insufficient to justify any attempt to state a general law. Recent experiments have been made by the author, Hopkinson, and Petavel to determine the rate at which a mass of heated gases, at from 1000° to 1600° C., loses heat to the comparatively cool metal surfaces which enclose it. These experiments will be described later, meantime it is sufficient to say that the rate of flow is rapid. Otherwise, it would be impossible to raise steam with the relatively small heating surfaces generally used in boilers. Before applying the efficiency values obtained to actual practice it is necessary to know at what rate a cubic foot of gases at about 1600° C. in contact with metal walls at from 17° C. to 100° C. will lose heat; also to know how that rate changes with change of temperature and density. Much is known of the laws of cooling at lower temperatures, but little positive data exist for temperatures so high as those occurring in the gas engine. A hot gas loses heat to the colder walls enclosing it mainly by circulation or convection. The conductivity of gases for heat is very slight, and unless in some way a large surface of the gas is exposed to the cooling surface, practically no heat would escape from the working fluid in the short time during which it is exposed in gas engines. Any arrangement which favours or hastens convection will therefore increase loss by increasing the extent of hot gaseous surface exposed to the walls. The smaller the surface to which a given volume of working fluid is exposed the less heat will it lose in a given time. So far as loss of heat is concerned, then, the best type of engine is that which exposes a given volume of working fluid to the smallest surface in performing its cycle. Suppose that in the three types the pistons move at the same velocity, then that which requires to move through the smallest volume, the areas of the pistons being supposed equal, will take the shortest time to perform its cycle. In the first engine the piston moves through 2·7 vols., with the hot air filling the cylinder; the second, through 3·7 vols.; and the third, through 2·4 vols. (see figs. 22, 24, and 26). As the volumes are proportional to the time taken to perform each cycle the third type has the best of it, the time of exposure of the hot working fluid being the least; the second type is worse than the first. There is still another circumstance in addition to surface exposed and time of exposure, that is, the average temperature of the hot gas which is exposed. If the average temperature is lower in one type than in another during exposure to a given surface for a certain time, then obviously less heat will be lost in the one than in the other. Comparing the average temperatures it is found that in the first the temperature ranges from 1600° C. to 817° C.; in the second from 1600° C. to 901° C.; and in the third from 1600° C. to 510° C. The third will therefore show a lower average temperature than the others. Three

conditions are requisite in the engine which is to lose the minimum of heat from its working fluid :

1. In performing its cycle it should expose a given volume of its working fluid to the least possible cooling surface ;
2. It should expose it for the shortest possible time ;
3. The average temperature during the time of exposure should be as low as possible—

which conditions are best fulfilled by the third type. In addition to its advantage in theoretic efficiency it possesses the further good points in practice of proportionally small cooling surfaces, short time of exposure, and rapid depression of temperature due to work done, consequently small loss of heat to the cylinder and piston.

The diagrams, figs. 22, 24, and 26, have been selected from the others belonging to each type because the pressures, temperatures, and relative volumes closely correspond with those which would be best and at the same time readily practicable.

The flow of heat really occurring in the gas engine cylinder will be discussed when the actual diagrams come under consideration ; meantime, it is sufficient to have proved that the third type will in practice give results more closely approaching its theory than the others. If in each case a constant proportion of the heat supplied were lost to the cylinder and piston, the ratio of the efficiencies would remain constant, and although it would be impossible from present data to predict the actual values, yet the relative values would be known.

Gain of Heat by the working Fluid when entering the Engine

In all types of gas engine it is found most economical to keep the motor cylinders as hot as possible ; they are generally worked at a temperature close upon the temperature of boiling water.[1] This is done to diminish the loss of heat from the explosion. It follows that if the working fluid is introduced at a lower temperature it becomes heated. In the first type, the charge should be admitted and remain at the lowest temperature until the moment of explosion, which is of course impossible if the cylinder is at 100° C. As the piston itself is hotter than that, it may be supposed that the charge is heated to that point.

Taking an extreme case and calculating the effect of having an absolute temperature of 390° for the lower limit, it will be found that the efficiency is diminished. In case 1, type 1, where the expansion is

[1] In large gas engines the water jackets are kept as cold as possible.

carried to atmosphere with a maximum temperature of 1873° absolute = 1600° C., the value becomes reduced to 0·23.

With a maximum temperature of 1273° absolute = 1000° C. the efficiency is 0·16.

TYPE I.

Initial temp. of working fluid	Max. temp.	Efficiency
17° C.	1600° C.	0·29
117° C.	1600° C.	0·23
17° C.	1000° C.	0·23
117° C.	1000° C.	0·16

Here heating while introducing the charge will always cause diminution in efficiency, the proportion of loss being greater with the lower maximum temperature. At 1600° C. the loss is nearly one-fifth, while at 1000° C. it is close upon one-fourth.

It is very difficult to say whether it is better to work with the cylinder hot or cold. The constructor finds himself in a dilemma; if the cylinder is kept as cold as the surrounding air, then the hot gases cool more rapidly. If he keeps the cylinder hot to diminish this, the efficiency falls also. Experiment alone can decide the question.

In engines of type 2 it is a usual proceeding to leave the compression cylinder entirely without water-jacketing, under the impression that heat is thereby saved; the temperature consequently rises to very nearly that of compression, and the entering charge becomes considerably heated before compression. This is especially the case if the admission area is small, and throttling occurs; all the energy of velocity of the entering gas becomes transformed into heat. As in the previous case the charge may be considered to rise to 117° C. before compression.

Where expansion is carried to atmosphere it has been shown that the efficiency is quite independent of the maximum temperature, but is determined by one circumstance only—the amount of the compression. As

$E = 1 - \frac{t}{t_c}$ and t is the temperature absolute before compressing

t_c ,, ,, ,, ,, after ,,

and as $\frac{t_c}{t} = \left(\frac{p_c}{p}\right)^{\frac{\gamma-1}{\gamma}}$, it follows that with a constant ratio between the pressures before and after compression, the ratio of temperature before and after compressing will also remain constant; that is, the efficiency is not in any way affected by heating the working fluid, provided the same degree of compression is used. Increase of

temperature previous to compression causes a proportional increase of temperature after compressing without in any way disturbing the ratio between them.

This is an important if in appearance a somewhat paradoxical fact, and it may be stated in another way :

If an engine receives all its supply of heat at one pressure, and rejects all its waste heat at another pressure, after falling from the higher to the lower pressure by expansion doing work, the efficiency is constant for all maximum temperatures of the working fluid.

The proportion of heat converted into work is not changed in any way by increasing the temperature before compressing, and if only one degree of heat be added after compressing, the same proportion of that one degree is converted into work, as would be done with any addition of heat however great.

Where the expansion is not continued enough to reduce the pressure after heating, to atmosphere, as in the cases of this type which occur in practice, this is not quite true ; the compression still remains the most powerful element of efficiency, but heating before compression produces some change, just as increase of temperature after compression produces change. The change is not great, and it is always in the direction of improvement with a limited expansion. If the lower temperature t is increased, the compression temperature t_c increases in proportion, and is accordingly nearer the maximum temperature. The volume increases less on heating, so that the effect upon efficiency is the same as if the expansion had been increased ; the terminal pressure will more closely approach atmosphere, and therefore come nearer to the condition of maximum efficiency.

In engines of type 3 the compression and expansion are often performed in the same cylinder. For this purpose it is necessary to leave at the end of the cylinder a space into which the charge is to be compressed. As the piston does not enter this space, a considerable volume of exhaust gases remains to mix with the fresh cold charge. Partly from this and partly from the heating effect of the cylinder and piston, the charge becomes considerably heated before compression. The temperature of 200° C. is not unusual. Here the simplest case is that where the expansion is continued to the same volume as existed before compression. The efficiency depends solely upon the amount of the compression ; for any given degree of compression it is constant, whether the addition of heat at constant volume after compression be great or small. The efficiency is $E = 1 - \frac{t}{t_c}$ as in type 2 (see p. 80) ; and the two absolute temperatures vary in the same ratio, that is, if the charge is heated before compression, the temperature after compression will be increased in the same ratio. The two temperatures

will therefore bear a constant ratio to each other, whatever the initial temperature may be, provided the compression is constant. Heating the charge before compression will consequently have no disturbing effect upon the theoretical efficiency.[1]

Where the expansion is carried to atmosphere the case is different. The diagram (fig. 26) may be considered to be made up of two parts giving two different efficiencies, the sum of which in this case is 0·51. In expanding from the compression volume v_c to the original volume v_o (compression 75 lbs. per sq. in.) the total efficiency is 0·37, and from that volume to v and atmospheric pressure, 0·14. The latter portion still obeys the same law as in a similar case of type 1; so that if the initial temperature at volume v_o be supposed 117° C. it will lose efficiency in a similar way. The temperature 901° C. will still exist at that point of the expanding line, so that it may be taken as similar to the case calculated on p. 106, where 1000° C. is the maximum. The loss of efficiency there is from 0·23 to 0·16 for an initial temperature of 117° C., which makes 0·14 become nearly 0·10. The total efficiency would therefore be 0·47 instead of 0·51 without previous heating.

Efficiency diminishes with increased temperature of working fluid before compressing if the expansion is carried to atmosphere, but does not change where the expansion is limited to the initial volume.

Other Causes of Loss

The third, fourth, fifth, sixth, seventh and eighth causes of loss require for their examination a comparison of the actual diagrams, and a knowledge of the phenomena of explosion and combustion, and so cannot be discussed at this stage.

[1] It is here necessary to distinguish between theoretical and practical efficiency. Heating before compression diminishes efficiency in practice by increasing maximum temperature, and therefore loss of heat.

CHAPTER V

COMBUSTION AND EXPLOSION

In the preceding chapters the gas engine has been considered simply as a heat engine using air as its working fluid; it has been assumed that in the different cycles the engineer is able to give the supply of heat either instantaneously, or slowly, at will; and also that he can command temperatures so high as 1000° C. or 1600° C. It is now necessary to study the properties of gaseous explosive mixtures in order to understand how far these assumptions are true.

On True Explosive Mixtures

When an inflammable gas is mixed with oxygen gas in certain proportions, the mixture is found to be explosive: a flame approached to even a small volume contained in a vessel open to the air will produce a sharp detonation. Variation of the proportions will cause change in the sharpness of the explosion. There is a point where the mixture is most explosive; at that point the inflammable gas and the oxygen are present in the quantities requisite for complete combination. After explosion the vessel will contain the product or products of combustion only, no inflammable gas remaining unconsumed, or oxygen uncombined, both having quite disappeared in forming new chemical compounds.

That mixture may be called the true explosive mixture.

Definition.—When an inflammable gas is mixed with oxygen in the proportion required for the complete combination of both gases, the mixture formed is the true explosive mixture.

If the chemical formula of an inflammable gas is known, the volume of oxygen necessary for the true explosive mixture can be at once calculated. Elementary substances combine chemically with each other in certain weights known as the atomic or combining weights: chemical symbols are always taken as representing those weights of the elements indicated. In dealing with inflammable gases used in the gas engine it is convenient to remember the following symbols and weights:

Element	Symbol	Combining weight
Oxygen	O	16
Hydrogen	H	1
Nitrogen	N	14
Carbon	C	12
Sulphur	S	32

In entering or leaving any compound the elements invariably enter or leave in weights proportional to those numbers or multiples of them. Thus hydrogen and oxygen combine with each other, forming water; the formula of the compound is H_2O, meaning that 18 parts by weight contain 16 parts of O and 2 parts of H. Similarly when carbon combines with oxygen two compounds may be formed, according to the conditions, carbonic oxide or carbonic acid, formulæ CO and CO_2, the former containing in 28 parts by weight, 12 parts of carbon and 16 parts of oxygen; the latter in 44 parts by weight containing 12 parts of carbon and 32 parts of oxygen.

The formula of a compound therefore not only indicates its nature qualitatively, but it also indicates its quantitative composition.

H_2O not only tells the nature of water, but it represents 18 parts by weight; CO means 28 parts by weight of carbonic oxide: CO_2 means 44 parts by weight of carbonic acid. The numbers 18, 28, and 44 are known as the molecular weights of the three compounds in question.

When dealing with gases it is more convenient to think in volumes than in weights. It is easier, for instance, to measure the proportions of explosive mixtures by volume and to say this mixture contains one cubic inch, one cubic foot, or one volume of inflammable gas to so many cubic inches, feet, or volumes of oxygen.

Fortunately there exists a simple relationship between the volumes of elementary gases and their combining weights, and also between the volumes of compounds and their molecular weights.

If equal volumes of the elementary gases are weighed, under similar conditions of temperature and pressure, it is found that their weights are proportional to the combining weights. Taking the weight of the hydrogen as 1, then the weights of equal volumes of nitrogen and oxygen are 14 and 16 respectively. If then it is wished to make a mixture of hydrogen and oxygen gases in the proportion of 2 parts by weight of the former to 16 parts by weight of the latter, it is only necessary to take 2 vols. H and 1 vol. O. The law may be stated in two ways, as follows:

Taking hydrogen as unity the specific gravity of the elementary gases is the same as their combining weights; or

The combining volumes of the elementary gases are equal.

Instead of troubling to weigh out portions of the gases it is at once known that one volume of nitrogen weighs 14 parts, the same volume of hydrogen weighing 1 part, oxygen 16 parts, and so on through all the gaseous elements, under the same temperatures and pressures.

Knowing that water is the compound formed by the combustion of hydrogen and oxygen, and that its formula is H_2O, it is at once apparent that the true explosive mixture of these gases is 2 vols. H. and 1 vol. O. By experiment it is found that the volume of the water produced is less (of course in the gaseous state) than the volume of the mixed gases before combination.

The measurement requires to be made at a temperature high enough to keep the steam formed in the gaseous state. Measure 2 vols. H and 1 vol. O into a strong glass vessel heated to 130° C.; the total is 3 vols.; fire by the electric spark over mercury. It will be found that the steam formed when it has cooled to 130° C. after the explosion measures 2 vols. It has been found to be true for all gaseous compounds, that however many volumes of elementary gases combine to form them the product is always two volumes. In elementary gases, one volume always contains the combining weight; in compound gases, two volumes always contain the molecular weight. Compared with hydrogen, therefore, the specific gravity of a gaseous compound is always one-half of the molecular weight.

As before, the law may be stated in two ways:

Taking hydrogen as unity, the specific gravity of a compound gas is half its molecular weight; or

The combining volume of a compound gas is always equal to double that of an elementary gas.

These laws are known as Gay-Lussac's laws, and form part of the very basis of modern chemistry.

Using them, the true explosive mixtures by volume and the volumes of the products of the combination can be found for any gas or mixture of gases, whether elementary or compound.

The inflammable compound gases, used in the gas engine, forming some of the constituents of coal gas are:

Inflammable gas	Formula	Molecular weight	Molecular vol.
Marsh gas	CH_4	16	2
Ethylene	C_2H_4	28	2
Carbonic oxide	CO	28	2

Applying Gay-Lussac's laws, the oxygen required for true explosive

mixtures and the volumes of the products of combustion are as follows for all the inflammable gases used in the gas engine :

	H_2O Steam.	CO_2 Carbonic acid.
2 vols. hydrogen (H) require 1 vol. oxygen (O) forming . .	2 vols.	—
2 vols. marsh gas (CH_4) require 4 vols. oxygen (O) forming .	4 vols.	2 vols.
2 vols. ethylene (C_2H_4) require 6 vols. oxygen (O) forming . .	4 vols.	4 vols.
2 vols. carbonic oxide (CO) require 1 vol. oxygen (O) forming .	—	2 vols.
2 vols. tetrylene (C_4H_8) require 12 vols. oxygen (O) forming .	8 vols.	8 vols.

With hydrogen and oxygen 3 volumes before combination become 2 volumes after combination. CH_4 and O, also C_2H_4 and O, the volumes of the products of combustion, are equal to the volumes of mixture. With carbonic oxide and oxygen 3 volumes before become 2 volumes after combination.

On Inflammability

Previous to 1817, Sir Humphry Davy made the admirable researches which led him to the invention of the safety lamp. He then made experiments upon different explosive mixtures, and found that under certain conditions they lost the capability of ignition by the electric spark. True explosive mixtures, he observed, may lose inflammability in two ways : by the addition of excess of either of the gases or of any inert gas such as nitrogen, and by rarefaction. The hydrogen explosive mixture, if reduced to one-eighteenth of ordinary atmospheric pressure, cannot be inflamed by the spark. Heated to dull redness at this pressure it will recover its inflammability and the spark will cause combination.

One volume of the mixture to which has been added nine volumes of oxygen is uninflammable, but if the density is increased or the termperature raised, it recovers its inflammability.

Eight volumes of hydrogen added, produces the same effect as the nine volumes of oxygen, but only one volume of marsh gas or half a volume of ethylene is required. The excess which destroys inflammability varies with the temperature, increasing with increase of temperature. Heating the mixture widens the range, both of dilution with excess or inert gas and reduction of pressure.

The point where inflammability ceases by diluting is very abrupt and sharply defined. The author has found that a coal gas which will inflame by the spark in a mixture of 1 gas and 14 air will not inflame with 15 of air. If the experiment be repeated on a warmer day, it may inflame with 15 of air but will not with 16 air. As the proportion is fixed for any given temperature it will be convenient to call that proportion for any mixture the 'critical proportion.' Any mixture in the critical proportion becomes inflammable by a very small increase of

temperature or pressure. The exact limits of dilution, temperature, and pressure have yet to be discovered.

Passing from any true explosive mixture by dilution to the mixture in the critical proportion, the inflammability slowly diminishes, the explosion becoming less and less violent, till at last no report whatever is produced, and the progress of the flame (if a glass tube be used) is easily followed by the eye.

In his great work on gas analysis Professor Bunsen confirms Davy's observations in every particular, proving loss of inflammability by dilution and reduction of pressure as well as its restoration by heating, increase of pressure and slight addition of the inflammable gas. His work, however, was not published till 1857.

On the Rate of Flame-Propagation

The sharp explosion of a true explosive mixture is due to the very rapid rate at which a flame, initiated at one point, travels through the entire mass and thereby causes the maximum pressure to be rapidly attained. With a diluted mixture the flame travels more slowly. Dilution therefore diminishes explosiveness in two ways—by increasing the time of getting the highest pressure and also by diminishing the highest pressure which can be got. Professor Bunsen's experiments are the earliest attempts to measure the velocity of flame movement in explosive mixtures. His method is as follows :

The explosive mixture is allowed to burn from a fine orifice of known diameter, and the rate of the current of the issuing gas carefully regulated by diminishing the pressure to the point at which the flame passes back through the orifice and inflames the explosive mixture below it. This passing back of the flame occurs when the velocity with which the gaseous mixtures issue from the orifice is inappreciably less than the velocity with which the inflammation of the upper layers of burning gas is propagated to the lower and unignited layers. Knowing then the volume of mixture passing through the orifice and its diameter, the rate of flow at the moment of back ignition is known. It is identical with the rate of flame-propagation through the mixture.

Bunsen made determinations for the true explosive mixtures of hydrogen and carbonic oxide.

Velocity of Flame in true Explosive Mixtures. (*Bunsen*)

Hydrogen mixture (2 vols. H and 1 vol. O) . .	34 metres per sec.
Carbonic oxide mixture (1 vol. CO and 1 vol. O).	1 metre per sec. nearly.

The method is a singularly simple and beautiful one, and answered thoroughly for Professor Bunsen's purpose at the time he devised it. Several objections, however, may be brought against it. The mixture in issuing from the jet into the air as flame, becomes mixed to some extent

with the air and so cools down; the metal plate also, pierced with the orifice, exercises a great cooling effect. If the hole were made small enough the flame could not pass back at all, however much the flow is reduced, because the heat would be conducted away so rapidly as to extinguish the flame. This had been shown by Davy in 1817; indeed it is the principle of the safety lamp. These causes probably make Bunsen's velocities too low. MM. Mallard and Le Chatelier have made velocity determinations by a method designed to obviate those sources of error.

The explosive mixture is contained in a long tube of considerable diameter, closed at one end, open to the atmosphere at the other. At each end a short rubber tube terminates in a cylindrical space closed by a flexible diaphragm. A light style is fixed upon the diaphragms. A drum revolves close to each style, both drums upon the same shaft. A tuning-fork, vibrating while the experiment is being made, traces a sinuous line upon the drum and so the rate of revolution is known. The mixture is ignited at the open end, and the flame in passing the lateral opening leading to the first diaphragm ignites the mixture there, and so moves the style and marks the drum; the arrival of the flame is signalled at the other end in the same way. The drums revolving together, the distance between the two style-markings measured by the vibration marks of the tuning-fork gives the time taken by the flame to move between the two points. The numbers got in this way are the rates of the communication of the flame through the mixture, back into the tube, while the flame can freely expand to the air; when both ends are closed the velocity is much greater. Then not only does the flame spread from particle to particle of the explosive mixture at the rate due to contact of the inflamed particles with the uninflamed ones, but the expansion produced by the inflammation projects the flame mechanically into the other part and so produces an ignition, which does not travel at a uniform rate, but at a continually accelerating one. In the same way, using the open tube but firing at the closed end, the expansion of the first portion adds to the apparent velocity of propagation, and projects the last portion of the mixture into the atmosphere. The true velocity of the propagation is the rate at which the flame proceeds from particle of inflamed mixture to uninflamed particle by simple contact; the true velocity depends upon inflammability alone, the rate under other conditions depends also upon heat evolved, and therefore movement due to expansion, mechanical disturbance of the unignited by the projection of the ignited portion into its midst. These conditions may vary much; the inflammability remains constant.

Mallard and Le Chatelier's results for the true velocity of propagations are:

VELOCITY OF FLAME IN TRUE EXPLOSIVE MIXTURES
(*Mallard and Le Chatelier*)

	Per sec.
Hydrogen mixture (2 vols. H and 1 vol. O)	20 metres
Carbonic oxide (2 vols. CO and 1 vol. O) . . .	2·2 ,,

Bunsen's rate for hydrogen mixture seems to have been too great, and for carbonic oxide mixture too little. The rate for a true and very explosive mixture such as hydrogen is liable to be inaccurately determined, as temperature variation makes a great change, and it is difficult even with Mallard and Le Chatelier's method to obtain concordant experiments. With less inflammable mixtures the difficulty disappears. As true explosive mixtures are never used in the gas engine, their properties concern the engineer only as a preliminary to study of diluted mixtures. The most explosive mixture which can be made with air contains a large volume of nitrogen inevitably present as diluent.

The following are some of their results with diluted mixtures, which are stated to be correct within 10 per cent. error of experiment :

VELOCITY OF FLAME IN DILUTED MIXTURES. (*Mallard and Le Chatelier*)

	Per sec.
1 vol. hydrogen mixture + $\frac{1}{3}$ vol. oxygen	17·3 metres
,, ,, + 1 vol. oxygen . . .	10 ,,
,, ,, + $\frac{1}{2}$ vol. hydrogen .	18 ,,
,, ,, + 1 vol. hydrogen	11·9 ,,
,, ,, + 2 vols. hydrogen . . .	8·1 ,,

These rates show that the true explosive mixture of hydrogen and oxygen when diluted with its own volume of oxygen falls from 20 metres per second to 10 metres, that is, it becomes one-half as inflammable ; when its own volume of hydrogen is the diluent, the velocity only falls to 11·9 metres per second. Hydrogen therefore has less effect in diminishing inflammability than oxygen.

Remembering the fact that the atmosphere contains one-fifth of its volume of oxygen, the remaining four-fifths being nearly all nitrogen, it is easy to get the proportions for the strongest explosive mixture possible with air. Two volumes hydrogen require 1 volume oxygen, and therefore 5 volumes air. The strongest possible mixture with air is two-sevenths hydrogen, five-sevenths air. The following experiments are for hydrogen and air in different proportions :

VELOCITY OF FLAME IN DILUTED MIXTURES. (*Mallard and Le Chatelier*)

	Per sec.
Mixture, 1 vol. H and 4 vols. air	2 metres
,, 1 ,, H and 3 vols. air	2·8 ,,
,, 1 ,, H and $2\frac{1}{3}$ vols. air	3·4 ,,
,, 1 ,, H and $1\frac{6}{7}$ vols. air	4·1 ,,
,, 1 ,, H and $1\frac{1}{2}$ vols. air	4·4 ,,
,, 1 ,, H and 1 vol. air	3·8 ,,
,, 1 ,, H and $\frac{1}{3}$ vol. air	2·3 ,,

Very strangely the velocity is greatest when there is an excess of hydrogen present. To get just enough of oxygen for complete burning, 1 volume H requires $2\frac{1}{2}$ volumes air, which would be naturally supposed to be the most inflammable mixture, as it gives out the greatest heat, but for some reason it is not. When the hydrogen is increased beyond 1 volume H to $1\frac{1}{2}$ volume air the velocity again falls off. A determination for coal gas and air gave 1 volume gas, 5 volumes air a velocity of 1·01 metre per second, and 1 volume gas, 6 volumes air 0·285 metre per second. With coal gas also the maximum velocity is got with the gas slightly in excess.

So far, these rates of ignition or inflammation are measures of inflammability, and are the rates for constant pressure; the rates for constant volume are very different, and the problem is a more complex one. Inflaming at the closed end of the tube, they found that even very dilute mixtures gave a sharp explosion, and in the case of hydrogen true explosive mixture, the velocity became 1000 metres per second instead of 20. With hydrogen and air 300 metres per second were obtained.

MM. Berthelot and Vieille have proved that under certain conditions even greater velocities than these are possible. The conditions, however, are abnormal, and the generation of M. Berthelot's explosive wave is exceedingly undesirable in a gas engine. It is generated by inflaming a considerable portion of the mixture at once, and so causing the transmission of a shock from molecule to molecule of the uninflamed mixture: this shock causes an ignition velocity nearly as rapid as the actual mean velocity of movement of the gaseous molecules at the high temperatures of combustion. The difference between this almost instantaneous detonation and the ordinary flame-propagation may be compared with similar differences in the explosion of gun-cotton discovered by Sir Frederic Abel. Gun-cotton lying loosely, and open to the air, will burn harmlessly if ignited by a flame; indeed, a considerable portion may be laid upon the open hand and ignited by a flame without the smallest danger. The same quantity in the same position, if fired by a percussive detonator, will occasion the most violent explosion, the nature of the shock given to the gun-cotton by the detonator causing a transmission of the kind of vibration necessary to cause its almost instantaneous resolution into its component gases.

The explosive wave in gases seems to originate in like conditions. Its velocity for the true explosive mixture of hydrogen and oxygen is 2841 metres per second, and for carbonic oxide mixture 1089 metres per second. The velocity is independent of pressure between half an atmosphere and one and a half atmosphere. It is independent, too, of the diameter of the tube used, within considerable limits, or of the

material of the tube, rubber and lead tubes giving similar results. Diluting the mixtures diminishes, and heating increases it. The experiments are very interesting and important, from a physicist's standpoint, but, fortunately for the inventor dealing with gas engines, the explosive wave is not easily generated in a gas engine cylinder ; if it were, it would be impossible to run the engines without shock and hammering.

The velocity which really concerns the engineer is that due to inflammability, and expansion produced by inflaming—the velocity, in fact, with which the inflammation spreads through a closed vessel. As it cannot be discussed without considering other matters—heat evolved by combustion, and temperatures and pressures produced—it will be advisable first to give the heat evolved by combustion, and then devote a complete chapter to explosion in a closed vessel.

Heat evolved by Combustion

Careful experiments upon the heat evolved by the combustion of gases in oxgyen have been made by Favre and Silberman, and also by Professor Andrews. The physicists first named burned the gases at constant pressure in a specially devised calorimeter. Professor Andrews mixed the gases in a thin spherical copper vessel, closed it, and exploded by the spark : the vessel being surrounded by water gave up its heat to the water, the weight of which being known, the rise of temperature gave the heat evolved.

Quantities of heat are measured by taking water as the unit. In this work, unless otherwise stated, a heat unit always means the amount of heat necessary to raise unit weight of water through 1° C.

Taking an average of Favre and Silberman and Andrews's results, the inflammable gases used in gas engines evolve upon complete combustion the following amounts of heat :

	Heat units
Unit weight of hydrogen completely burned to H_2O evolves . .	34,170
Unit weight of carbon completely burned to CO_2 evolves. . .	8,000
Unit weight of carbonic oxide completely burned to CO_2 evolves .	2,400
Unit weight of marsh gas completely burned to CO_2 and H_2O evolves	13,080
Unit weight of ethylene completely burned to CO_2 and H_2O evolves	11,900

That is, one pound weight of hydrogen burned completely to water will evolve as much heat as would raise 34,170 lb. of water through 1° C., or the converse. One pound of carbon in burning to carbonic acid evolves as much heat as would raise 8000 lb. of water through 1° C. These numbers give the amount or quantity of heat evolved. The intensity or temperature of the combustion may be calculated on the assumption that the whole heat is evolved under such conditions that no heat is lost, or is applied to anything else but the products of

combustion. To make the calculation it is necessary to know the specific heat of the products.

The amount of heat required to heat unit weight of water through one degree is 1 heat unit, the specific heat of any other body is the number of heat units required to heat unit weight of the body through one degree. Gases have two different specific heats depending upon whether heat is applied while the gas is kept at constant volume, or at constant pressure; both are required in dealing with gas engine problems. The specific heat at constant volume is sometimes known as the true specific heat; in taking the specific heat at constant pressure the gas necessarily expands, and so does work on the external air; this specific heat is therefore greater than the former by the amount of work done. For the gases used in the gas engine the two values are as follows. The ratio between the two is also given, as it is frequently required in efficiency calculations. The experimental numbers are Regnault's, the calculated specific heat at constant volume, Clausius.

SPECIFIC HEATS OF GASES TO 200° C.

Regnault

(*For equal weights. Water* = 1)

Name of gas	Sp. heat at constant pressure	Sp. heat at constant volume	$\frac{\text{Sp. heat con. pres.}}{\text{Sp. heat con. vol.}}$
Air	0·237	0·168	1·408
Oxygen	0·217	0·155	1·403
Nitrogen	0·244	0·173	1·409
Hydrogen	3·409	2·406	1·417
Marsh gas	0·593	0·467	—
Ethylene	0·404	0·332	1·144
Carbonic oxide	0·245	0·173	1·416
Steam	0·480	0·369	1·302
Carbonic acid	0·216	0·171	1·165

It is convenient to remember that the specific heats of combining or atomic weights of the elements are equal—Dulong and Petit's law. To this law there are few exceptions, and the permanent elementary gases, oxygen, nitrogen, and hydrogen, obey it almost absolutely. As equal volumes of these gases represent the combining weights, it follows that equal volumes of these gases have the same specific heat. Taking the specific heat of air as the unit, the specific heat of hydrogen and oxygen gases is also unity.

The compound gases do not obey the law so closely.

Regnault's experimental method determined only the specific heat at constant pressure K_p, and the specific heatat constant volume K_v is calculated from these numbers.

Regnault also studied the effect of changing temperature on the specific heat of air and carbonic acid, and he concluded that within his experimental temperature range the specific heat of air remained constant, but that of carbonic acid increased materially. This is shown by the following table:

SPECIFIC HEAT AT CONSTANT PRESSURE WITH VARYING TEMPERATURES FOR AIR AND CARBONIC ACID

Regnault

Air		Carbonic Acid	
Temperature, ° C.	Sp. heat K_p	Temperature, ° C.	Sp. heat K_p
From − 31 to + 10	0·2377	From − 30 to + 10	0·1843
0 to + 100	0·2374	+ 10 to + 100	0·2025
0 to + 200	0·2375	+ 10 to + 210	0·2169

Air obviously remains constant from − 31° to + 200° C., only the fourth decimal changes; carbonic acid, however, changes from 0·184 to 0·217, taking the nearest third decimal by nearly 18 per cent. in 200° C.

Dr. Joly was the first to determine by a direct calorimetric method, his well-known steam calorimeter, the specific heat of gases at constant volume K_v; he found that the specific heat varied to some extent with change of density. In both air and carbonic acid he found increase of specific heat K_v with increased density, but with hydrogen specific heat appeared to decrease with increase of density. For air the specific heat at constant volume at a mean pressure of 19·51 atmospheres and a mean density of 0·0205 was found to be 0·1721.

Carbon dioxide gave the following results:

SPECIFIC HEAT AT CONSTANT VOLUME WITH VARYING PRESSURES

Joly

Pressure in atmospheres	Density	K_v
7·20	0·011530	0·1684
12·20	0·019950	0·1705
16·87	0·028498	0·1714
20·90	0·036529	0·1730
21·66	0·037802	0·1738

The density in these figures is really specific gravity, water = 1.

Similar experiments with hydrogen proved that its mean specific heat at constant volume K_v was 2·402.

Comparing these numbers found experimentally with the K_v values calculated from Regnault's K_p results at pressure over atmosphere, it is seen that for air the value K_v at atmospheric pressure is 0·168, but at a mean pressure of 19·5 atmospheres it becomes 0·172; it increases by 2·3 per cent.

For carbonic acid Regnault's number $K_v = 0{\cdot}171$, Joly's maximum value is 0·1738; it is higher by 1·7 per cent.; but Joly's numbers were taken at the lower temperature of 100° C., so that they agree substantially with Regnault's values as given at p. 118. It is to be noted, however, that Joly's table shows that an increase from 7·2 to 21·6 atmospheres increases K_v from 0·168 to 0·174, or 3·6 per cent. Joly's value K_v for hydrogen is 2·402, which is in practical agreement with Regnault's 2·406.

Experiments were made by Mallard and Le Chatelier and by Berthelot upon gaseous explosions in a closed vessel at atmospheric initial pressure, from which they deduced values of K_v for temperatures up to and over 2000° C. for air, steam, carbonic acid and other gases, and these values indicated a large increase of specific heat for all the gases; the present writer, however, with many others found it impossible to accept their determinations, as he felt that no proof had been advanced that combustion had ceased before the measurements were made. He felt that no reliable determinations could be made without avoiding combustion altogether as the means of heating the gases where specific heat was to be found.

Messrs. Holborn and Austin have made a series of valuable experiments at the Reichsanstalt, Berlin, and at the British Association meeting of 1907 they gave the following tables for steam, carbonic acid, and nitrogen at temperatures from 110° C. to 1400° C. The experiments were made at constant atmospheric pressure like Regnault's, the gases were heated electrically, the temperature measured by a thermocouple, and the heat quantity by a calorimeter.

MEAN SPECIFIC HEATS AT CONSTANT PRESSURE WITH VARYING TEMPERATURES FROM 110° TO 1400° C. FOR STEAM, CARBONIC ACID, AND NITROGEN

Holborn and Austin

Temperature, ° C.	Steam H_2O	Temperature, ° C.	Carbonic acid CO_2	Nitrogen N
110 to 280	0·465	110 to 200	0·217	0·240
110 to 400	0·467	110 to 400	0·229	0·242
110 to 600	0·473	110 to 600	0·240	0·247
110 to 800	0·482	110 to 800	0·250	0·251
110 to 1000	0·494	110 to 1000	0·258	0·254
110 to 1200	0·510	110 to 1200	0·264	0·258
110 to 1400	0·532	110 to 1400	0·272	0·262

From these experiments it is proved that the mean specific heat of steam at constant pressure from 110° to 1400° is greater than that between 110° and 280° C. by nearly 14·5 per cent.; while with carbonic acid between 110° and 1400° C. and 110° and 200° the increase is 25·3 per cent. nearly. With nitrogen for the same temperature ranges as carbonic acid the increase is much smaller—it is only about 9 per cent.

The value of γ for steam at mean $K_p = 0·532$ is 1·26, so that for that temperature $K_v = \frac{0·532}{1·26} = 0·422$.

For carbonic acid having mean K_p to $1400° = 0·272$ the value of $\gamma = 1·21$ and

$$K_v = \frac{0·272}{1·21} = 0·225.$$

Comparing with Regnault's tables these values show very substantial increases for H_2O and CO_2, and a small but appreciable increase for N.

Earlier experiments of Holborn and Austin, using the same method, gave the corresponding values for air, nitrogen, and oxygen as follows:

MEAN SPECIFIC HEAT AT CONSTANT PRESSURE WITH VARYING TEMPERATURES FROM 20° TO 800° C FOR AIR, NITROGEN, AND OXYGEN

Holborn and Austin

Temperature, ° C.	Air	Nitrogen	Oxygen
20 to 440	0·2366	0·2419	0·2240
20 to 630	0·2429	0·2464	0·2300
20 to 800	0·2430	0·2497	—

These values agree substantially with the later determinations.

Holborn and Austin have calculated values of c_p at temperatures up to 800° as follows for CO_2, and they show corresponding values calculated from the Mallard and Le Chatelier experiments, as also from the later experiments of Langen.

C_p AT T° C FOR CO_2

T° C	Holborn and Austin C_p at T°	Langen C_p at T°	Mallard and Le Chatelier C_p at T°
0	0·2028	0·1980	0·1880
100	0·2161	0·2100	0·2140
200	0·2285	0·2220	0·2390
400	0·2502	0·2450	0·2840
600	0·2678	0·2690	0·3230
800	0·2815	0·2920	0·3550

Langen's experiments were made by the explosion pressure method used by Mallard and Le Chatelier, but a very large volume of gaseous mixture was employed, and the means of measuring the rise and fall of pressure was much improved compared with the early apparatus. Mallard and Le Chatelier's value for 800° C. is nearly 26 per cent. above that of Holborn and Austin, while Langen's figure is only about 3·5 per cent. higher. Langen's numbers closely approximate to Holborn and Austin's, while Le Chatelier's depart considerably on both sides.

The calculation of temperature of combustion can now be made.

The amount of heat evolved from unit weight of a combustible is usually said to measure its calorific power, that amount divided by the specific heat of the products of the combustion is said to be the measure of its calorific intensity. The calorific intensity is indeed the theoretical temperature of the combustion: taking hydrogen first, unit weight evolves 34,170 heat units. But the water formed weighs 9 units (from formula H_2O), and if its specific heat in the gaseous state were unity, the supposed maximum temperature of combustion would be $\frac{34{,}170}{9} = 3796{\cdot}6$. But the specific heat is less than unity; therefore the theoretical maximum will be greater. It is $\frac{34{,}170}{9 \times 0{\cdot}480} = 7909{\cdot}7$. For certain reasons to be considered later, no such enormous temperatures are ever attained by combustion. In the above calculation the latent heat of steam should first have been deducted, as it is included in the total heat evolved as measured by the calorimeter: it is 537 heat units. $34{,}170 - 537$ gives the total heat available for increasing the temperature; the amended calculation is $\frac{34{,}170 - 537}{9 \times 0{\cdot}480} = 7785{\cdot}4$, still an exceedingly high temperature.

Here we have taken Regnault's values as given on p. 118 for low temperatures. It is obvious that the real specific heat of steam through the longer temperature range must be greater. Assume c_p to be 0·532, the highest of Holborn and Austin's values, see p. 120, then the calculation becomes $\frac{34{,}170 - 537}{9 \times 0{\cdot}532} = 7024{\cdot}4$. This is still a very high temperature which has not been realised, so far as we know, by combustion or by any other method.

Calculating the heat evolved by burning carbon in the same way, but omitting any deduction for the latent heat of carbonic acid (it does not affect the calorimeter, as it does not condense), the theoretical temperature produced by burning in oxygen is still higher, being 10,174° C. Burning in air the theoretical temperatures are lower, as

the nitrogen present acts as a diluent, and must necessarily be heated to the same temperature as the products of the combustion. They are given as follows in 'Watts' Dictionary':

	Calorific power	Temperature produced °C.	
		In oxygen	In air
Carbon	8,080	10,174	2,710
Hydrogen	34,462	6,930	2,741

These are the supposed temperatures burning in the open atmosphere, and therefore at constant pressure, the gases expanding doing work upon the air. At constant volume, that is, burning in a closed vessel so that the volume cannot increase but only the pressure, the temperature should be greater as the specific heat at constant volume is less. Allowing for that, the numbers become

THEORETICAL TEMPERATURES OF COMBUSTION AT CONSTANT VOLUME

	Temperature produced ° C.	
	In oxygen	In air
Carbon	12,820	—
Hydrogen	9,010	4,119

Using the values given on p. 118 from Regnault and the thermal values at p. 117, we get 12 parts by weight of carbon burning in oxygen to CO_2 evolves $12 \times 8000 = 96{,}000$ Centigrade heat units: this produces 44 parts by weight of CO_2, having a c_p value of 0·216.

$$\frac{12 \times 8000}{44 \times 0.216} = 10{,}101^\circ \text{ C.}$$

Taking Holborn and Austin's highest value of 0·272 for c_p of CO_2, we get $\dfrac{12 \times 8000}{44 \times 0{\cdot}272} = 8021^\circ$ C.

For combustion of carbon in air, using Holborn and Austin's values, we get temperature of combustion in air 2321° C. For hydrogen in air, with the same number, we get 2776° C.

Such temperatures as 4000° C. have never been produced by combustion, for many reasons, of which all save the most potent have been discussed by the earlier writers on heat. This is Dissociation.

DISSOCIATION

Most chemical combinations, while in the act of formation from their constituent elements, evolve heat, and, as a general rule, the greater the heat evolved the more stable is the compound formed. The compound after formation may generally be decomposed by heating to a high enough temperature, heat being one of the most powerful splitting-up agencies known to the chemist. The nature of

the decomposition varies with the compound. In many cases the process is irreversible, that is, although heating up will cause decomposition, cooling down again, however slowly, will not cause recombination. In some compounds, however, under certain conditions the process is reversible, and recombination occurs on slow cooling.

Definition.—Dissociation may be defined as a chemical decomposition by the agency of heat, occurring under such conditions that upon lowering the temperature the constituents recombine.

Groves found long ago that water begins to split up into oxygen and hydrogen gases at temperatures low compared with that produced by combustion. Deville made a careful study of the phenomena, and found that decomposition commences at 960° to 1000° C. and proceeds to a limited extent: raising the temperature to 1200° C. increases it, but a limit is reached. The amount of decomposition depending upon the temperature, for each temperature there is a certain proportion between the amount of steam and the amount of free oxygen and hydrogen gases present. If the temperature is increased, the proportion of free gases also increases: if temperature is diminished, the proportion of free gases diminishes. If the temperature be raised beyond a certain intensity, the water is completely decomposed: if lowered beyond a certain temperature, complete combination results. The same thing happens with carbonic acid, the temperature of decomposition is lower.

It is quite evident, then, that at the highest temperatures produced by combustion, the product cannot exist in the state of complete combination. It will be mixed to a certain extent with the free constituents which cannot combine further until the temperature falls; as the temperature falls, combustion will continue till all the free gases are combined. The subject, from its nature, is a difficult one in experiment, and accordingly different observers do not agree upon temperatures and percentages of dissociation, but all are agreed that dissociation places a rigid barrier in the way of combustion at high temperatures, and prevents the attainment of temperatures, by combustion, which are otherwise quite possible. With no dissociation, hydrogen burning in oxygen should be able under favourable circumstances to give a temperature of over 6000° C., as has been shown. Deville's experiments upon the temperature of the oxyhydrogen flame, at constant pressure of the atmosphere, gave under 2500° C. The estimate was made by melting platinum in a lime crucible, with the oxyhydrogen flame playing upon the platinum, the crucible being well protected against loss of heat by lime blocks, so that the platinum could really attain the temperature of the flame; when at the highest temperature, the molten platinum was rapidly poured into a weighed calorimeter, and the rise in temperature noted. From this was

calculated the temperature of the platinum. The experiment was dangerous and inaccurate, but it is the only serious attempt which has been made to determine the temperature of the oxyhydrogen flame at constant pressure.

The highest temperature produced by hydrogen burning in oxygen has been determined by Bunsen, and also Mallard and Le Chatelier, for combustion at constant volume, that is, explosion.

As the theoretic calculation shows, with no dissociation a temperature of 9000° C. is possible. The highest maximum it is possible to assume from Bunsen's experiments is 3800° C. ; from Mallard and Le Chatelier's, 3500° C. The two sets of experiments are concordant. It is true the latter physicists do not attribute the difference wholly to dissociation, but they agree that part is due to this cause ; and that there is an enormous difference between the temperature actually got and that which should be possible if no limit existed, all are agreed. With air, Bunsen's figures show a maximum of about 2000° C., Mallard and Le Chatelier say 1830° C. ; the present writer has also made experiments with hydrogen in air, and finds the highest possible temperature to be 1900° C. The calculated maximum is 4119° C. The difference is not so great as with the true explosive mixture, which is to be expected, but all experiments agree in proving that there is a considerable difference.

CHAPTER VI

EXPLOSION AND COOLING IN A CLOSED VESSEL—EXPERIMENTAL INVESTIGATIONS

THE value of any inflammable gas for the production of power by explosion can be determined apart altogether from theoretical considerations by direct experiment. It is evident that the gas which for a given volume causes the greatest increase in pressure will give the greatest power for every cubic foot used, provided that the pressure does not fall so suddenly that it is gone before it can be utilised by the piston.

Two qualities will be possessed by the best explosive mixture: (1) greatest pressure per unit volume of gas; (2) longest time of maximum pressure when exposed to cooling.

In the gas engine itself the conditions are so complex that the problem is best studied in the first instance under simplified conditions. The author has made a set of experiments upon many samples of coal gas mixed with air in varying proportions, to find the pressures produced, and the duration of those pressures; igniting mixtures at atmospheric pressures and temperature, and also at higher temperature and initial pressures. He has made some experiments upon pure hydrogen and air mixtures in the same apparatus for comparison.

The experimental apparatus used by him in the years 1884 and 1885 is shown at fig. 35. It consists of a closed cylindrical vessel 7 ins. diameter and $8\frac{1}{4}$ ins. long, internal measurement, and therefore of 317 cubic ins. capacity. It is truly bored, and the end-covers turned so that the internal surface is similar to that of an engine cylinder; the covers are bolted strongly so as to withstand high pressures. Upon the upper cover is placed a Richards indicator, in which the reciprocating drum has been replaced by a revolving one; the rate of revolution is adjusted by a small fan, a weight and gear giving the power.

The cylinder is filled with the explosive mixture to be tested; the drum is set revolving, the pencil of the indicator pressed gently against it, and the electric spark is passed between the points placed at the bottom of the space. The drum is enamelled and the pencil is a

blacklead one. The pressure of the explosion acts upon the indicator piston, and a line is traced upon the drum, which shows the rise and fall of pressure. The rising line traces the progress of the explosion; the falling line the progress of the loss of pressure by cooling. The rate of the revolution of the drum being known, the interval of time elapsing between any two points of the explosion or cooling curve is also known. That is, the curve shows the maximum pressure attained, the time of attaining it, and the time of cooling. Line *b* on fig. 36 is a facsimile of the curve produced by the explosion of a

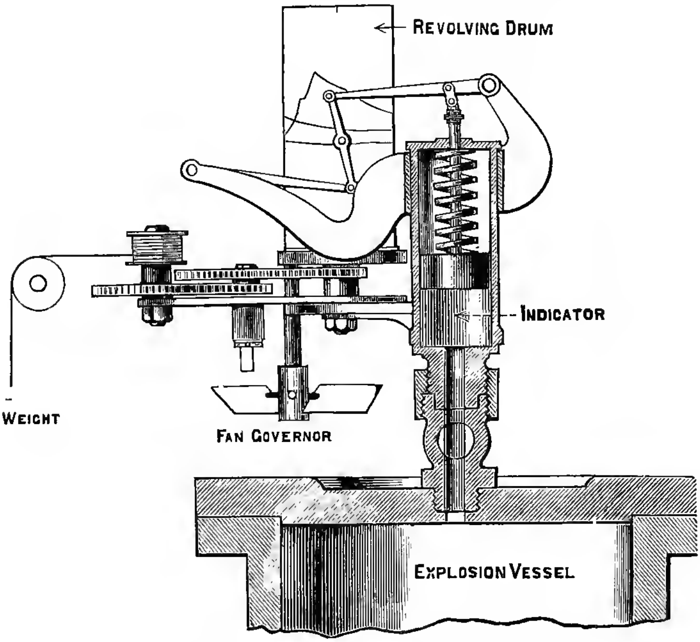
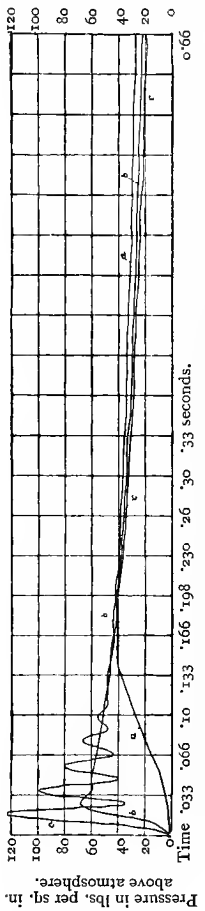


FIG. 35.—Clerk Explosion Apparatus of years 1884 and 1885

mixture containing 1 vol. hydrogen and 4 vols. air. Each revolution of the drum was accomplished in 0·33 sec., so that each tenth of a revolution takes 0·033 sec. The vertical divisions give time; the horizontal, pressures. In this experiment the maximum pressure produced by the explosion is 68 lbs. per sq. in. above atmosphere, and it is attained in 0·026 second. Compared with the rate of increase the subsequent fall is very slow. The rise occurs in 0·026 second; the fall to atmosphere again takes 1·05 second, or nearly sixty times the other. It is in fact an indicator diagram from an explosion where the volume is constant, the motor piston being absent, and the only

cause of loss of pressure is cooling by the enclosing walls. The exact composition of the mixture, its uniform admixture, the temperature and pressure before ignition, are all accurately known. After studying explosions under these known conditions, it becomes easier to understand what occurs under more complex conditions, where the moving piston makes the cooling surface change, and where the expansion doing work also requires consideration. As the rapidity of the increase of pressure measures the explosiveness of a mixture, the time occupied from the commencement of increase to maximum pressure will be called the *time of explosion*. The explosion is complete when maximum pressure is attained. It does not follow from this that the combustion is complete; that is another matter. The explosion arises from the rapid spreading of the flame throughout the whole mass of the mixture, which may be called the inflammation of the mixture. More or less rapid inflammation means more or less explosive effect, but not complete combustion. The complete burning of the gases present may not occur till long after complete inflammation.



Vessel used 7 ins. diam. and 8¼ ins. long. Scale of indicator spring, 1 lb. = $\frac{1}{14}$ in. Mixtures used: pure hydrogen and air—Experiment *a*, 1 vol. hydrogen to 6 vols. air; *b*, 1 to 4; *c*, 1 to 5. Temperature of gases before ignition, 16° C.; pressure (atmospheric), 14·7 lbs.

FIG. 36.—Explosion of Gaseous Mixtures. Experiments in a closed vessel.

The terms *combustion*, *explosion*, and *inflammation* will be used in this sense alone:

Combustion burning; complete combustion, the complete burning of the carbon of the combustible gas to carbonic acid, and the hydrogen to water. So long as any portion of the combustible remains uncombined with oxygen the combustion is incomplete.

Complete explosion, the attainment of maximum pressure.

Time of explosion; the time elapsing between beginning of increase and maximum pressure.

Complete inflammation, the complete spreading of the flame throughout the mass of the mixture.

Confusion has arisen through the indifferent use of these terms, which are really distinct and are not synonymous.

With mixtures made with Glasgow coal gas the author has obtained the following maximum pressures and times of explosion :

EXPLOSION IN A CLOSED VESSEL. (*Clerk*)

Mixtures of air and Glasgow coal gas

Temp. before explosion . . . . 18° C.
Pressure before explosion . . . . atmospheric.

Mixture.		Max. press. above atmos. in pounds per sq. in.	Time of explosion
Gas.	Air.		
1 vol.	13 vols.	52	0·28 sec.
1 vol.	11 vols.	63	0·18 sec.
1 vol.	9 vols.	69	0·13 sec.
1 vol.	7 vols.	89	0·07 sec.
1 vol.	5 vols.	96	0·05 sec.

The highest pressure which any mixture of coal gas and air is capable of producing without compression is only 96 lbs. per sq. in. above atmosphere, and the most rapid increase is not more rapid than always occurs in a steam cylinder at admission. Many are still prejudiced against gas, compared with steam, because of the so-called explosive effect, and the fear that gas explosions may occasion pressures quite beyond control, like solid explosives. The fear is quite unfounded ; the pressure produced by the strongest possible mixture of coal gas and air is strictly limited by the pressure before ignition, and can always be accurately known ; and so provided for by a proper margin of safety in the cylinders and other parts subject to it.

The most dilute mixture of air and Glasgow gas which can be ignited at atmospheric pressure and temperature contains $\frac{1}{14}$ of its volume of gas, and the pressure produced is 52 lbs. above atmosphere. The time of explosion is 0·28 second ; so slow is the rise that it cannot with justice be termed an explosion. It is too slow to be of any use in an engine running at any reasonable speed ; the stroke would be almost complete before the pressure had risen. The mixture containing $\frac{1}{6}$ of its volume of gas is that with just enough oxygen to burn the gas. It is anomalous that the highest pressure is given with excess of coal gas. The rate of ignition also is greatest with that mixture. This agrees with the results obtained by Mallard and Le Chatelier, excess of hydrogen giving the highest rate of inflammation.

Similar experiments were made with air and Oldham coal gas.

EXPLOSION IN A CLOSED VESSEL. (*Clerk*)

Mixtures of air and Oldham coal gas

Temp. before explosion . . . . 17° C.
Pressure before explosion . . . . atmospheric

Mixture		Max. press. above atmos. in pounds per sq. in.	Time of explosion
Gas	Air		
1 vol.	14 vols.	40	0·45 sec.
1 vol.	13 vols.	51·5	0·31 sec.
1 vol.	12 vols.	60	0·24 sec.
1 vol.	11 vols.	61	0·17 sec.
1 vol.	9 vols.	78	0·08 sec.
1 vol.	7 vols.	87	0·06 sec.
1 vol.	6 vols.	90	0·04 sec.
1 vol.	5 vols.	91	0·055 sec.
1 vol.	4 vols.	80	0·16 sec.

The highest pressure in this case is 91 lbs. per sq. in. above atmosphere, but the more rapid explosion is 0·04 second and 90 lbs. pressure, a little less pressure than is given by Glasgow gas but a slightly more rapid ignition. The mixtures are evidently more inflammable, as the critical mixture is $\frac{1}{15}$ volume of gas instead of $\frac{1}{14}$ as with Glasgow gas. Although repeatedly tried, a mixture of 1 volume gas 15 volumes air failed to inflame with the spark.

Hydrogen and air mixtures were also tested as follows :

EXPLOSION IN A CLOSED VESSEL. (*Clerk*)

Mixtures of air and hydrogen

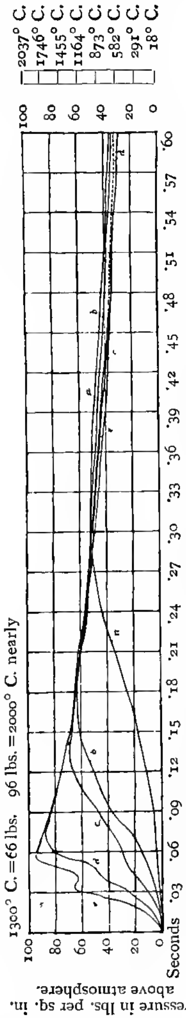
Temp. before explosion . . . . . . 16° C.
Pressure before explosion . . . . . . atmospheric

Mixture		Max. press. above atmos. in pounds per sq. in.	Time of explosion
Hyd.	Air.		
1 vol.	6 vols.	41	0·15 sec.
1 vol.	4 vols.	68	0·026 sec.
2 vols.	5 vols.	80	0·01 sec.

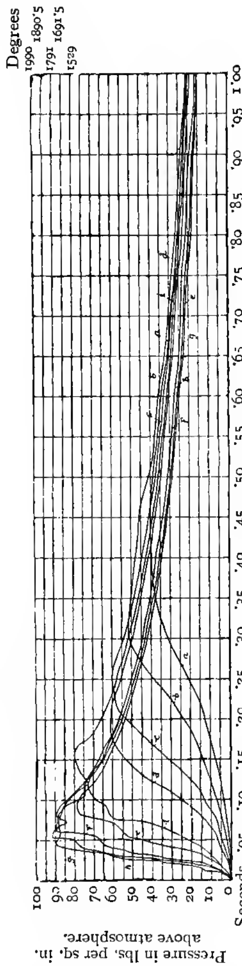
The inferiority of hydrogen to coal gas, volume for volume, is very evident ; the highest pressure is only 80 lbs. above atmosphere, and the mixture requires $\frac{2}{5}$ of its volume of hydrogen to give it, while coal gas gives the same pressure with about $\frac{1}{10}$ volume. The hydrogen mixture, too, ignites so rapidly that it would occasion shock in practice, the strongest mixture having an explosion time of one-hundredth of a second. With gas the most rapid is four-hundredths of a second.

The Best Mixture for use in Non-compression Engines

From these tables can be ascertained the best gas and the best mixture for use in non-compression engines with cylinders kept cold.



Vessel used 7 ins. diam. and 8¼ ins. long. Scale of indicator spring 1 lb. = $\frac{1}{48}$ in.
Mixtures used, Glasgow coal gas and air. Experiment *a*, 1 vol. gas to 13 vols. air; *b*, 1 to 11; *c*, 1 to 9; *d*, 1 to 7; *e*, 1 to 5.
Temperature of gases before ignition, 18° C.; pressure atmospheric.



Vessel used 7 ins. diam. and 8¼ ins. long. Scale of indicator spring, 1 lb. = $\frac{1}{32}$ in.
Mixtures used, Oldham coal gas and air. Experiment *a*, 1 vol. gas to 14 vols. air, temp. (of explosion and measuring vessels) 18° C.; *b*, 1 to 13, 16·2°; *c*, 1 to 12, 16°; *d*, 1 to 11, 16·1°; *e*, 1 to 9, 16·7°; *f*, 1 to 7, 16·7°; *g*, 1 to 6, 16°; *h*, 1 to 5, 16°; *i*, 1 to 4, 19°.
Centigrade temperatures, assuming no loss of volume, owing to combination of mixtures. Atmospheric pressure, 14·7 lbs., and mean temperature before ignition, 17° C.

Fig. 37.—Explosion of Gaseous Mixtures. Experiments in a closed vessel

Take first Glasgow gas, and determine which mixture gives the best result.

(1) Power of producing pressure.

Suppose one cubic inch of Glasgow coal gas to be used in each of the five mixtures, whose maximum pressures and times of explosion are given in the table on p. 129, the mixtures would measure respectively 14, 12, 10, 8, and 6 cubic ins. Let them be placed in cylinders of 14, 12, 10, 8, and 6 sq. in. piston area; the piston will in each case be raised one inch from the bottom of its cylinder. If the pressures upon the piston were the same, equal movements of piston would give equal power; if therefore the mixtures gave equally good results the maximum pressure multiplied by the piston area will in all cases be the same.

Multiplying 14, 12, 10, 8, and 6 by their corresponding pressures 52, 63, 69, 89, and 96 respectively, the products are 728, 756, 690, 712, and 576. These numbers are the pressures in pounds which each mixture is capable of producing with one cubic inch of Glasgow coal gas, cylinders of such area being used that the depth of mixture is in every case one inch.

Proportion of Glasgow gas in mixture .	$\frac{1}{14}$, $\frac{1}{12}$, $\frac{1}{10}$, $\frac{1}{8}$, $\frac{1}{6}$.
Pressure produced upon pistons by one cubic inch	728, 756, 690, 712, 576 lbs.

The best mixture is seen at a glance; it is that containing one-twelfth of gas. The pressure produced by one cubic inch of gas is at its highest value 756 lbs., in a cylinder of 12 ins. piston area, and containing 12 cubic ins. of mixture.

In modern gas engines the time taken by the piston to make the working part of its stroke is generally about one-fifth of a second. If the pressure in one mixture has fallen more, proportionally, in that time, then, although it may give the highest maximum, it may lose too rapidly to give the highest mean pressure. To find this cooling effect, find the pressure to which each mixture falls at the end of 0·2 second after maximum pressure; it is in the different cases:

Mixture containing gas	$\frac{1}{14}$, $\frac{1}{12}$, $\frac{1}{10}$, $\frac{1}{8}$, $\frac{1}{6}$.
Time after beginning explosion (0·2 sec. after max. pressure)	0·48, 0·38, 0·33, 0·27, 0·25 sec.
Pressure in lbs. per sq. in. . . .	43, 48, 47, 55, 57.
Press. × respectively by 14, 12, 10, 8, and 6	602, 576, 470, 440, 342.

The lower row expresses the relative pressures still remaining after allowing each explosion to cool for one-fifth of a second from complete explosion; they express the resistance to cooling possessed by the mixtures. It is evident at once that the strongest mixtures cool most rapidly; a higher temperature being produced, more of the heat of the explosion is lost in a given time.

(2) Power of producing pressure and resisting cooling.

To find the best mixture for producing pressure and resisting cooling, those numbers are to be added to the corresponding ones for maximum pressure :

Proportion of Glasgow gas in mixture .	$\frac{1}{14}$, $\frac{1}{12}$, $\frac{1}{10}$, $\frac{1}{8}$, $\frac{1}{6}$.
Pressure produced upon pistons by one cubic inch gas	728, 756, 690, 712, 576.
Pressure remaining upon pistons 0·2 sec. after complete explosion. . . .	602, 576, 470, 440, 342.
Mean pressure	665, 666, 580, 576, 459.

The mean of the two sets gives numbers expressing the relative values of the mixture for producing pressure, and at the same time resisting cooling. The two weakest mixtures are best in both respects; the low result given by the strongest mixture is due to the fact that excess of gas is present and it remains unburned, it proves how easily the consumption of an engine may be increased by even a slight excess of gas in the mixture.

The two best mixtures ignite too slowly, but in the actual engine that is easily controlled, as will be explained later. The best mixtures are 1 volume gas 13 volumes air, and 1 volume gas 11 volumes air. With more gas the economy will rapidly diminish.

The experiments with Oldham gas treated in the same way give the following results :

Proportion of Oldham gas in mixture	$\frac{1}{15}$, $\frac{1}{14}$, $\frac{1}{13}$, $\frac{1}{12}$, $\frac{1}{10}$, $\frac{1}{8}$, $\frac{1}{7}$, $\frac{1}{6}$, $\frac{1}{5}$.
Pressure produced upon pistons by one cubic inch gas . .	600, 721, 780, 732, 780, 696, 630, 546, 400.
Pressure remaining upon pistons 0·2 sec. after complete explosion per sq. in.	31, 40, 42, 44, 44, 47, 52, 50, 46.
Pressure per piston . . .	465, 560, 546, 528, 440, 376, 364, 300, 230.
Mean pressure upon piston . .	532, 640, 663, 630, 610, 536, 497, 423, 315.

Here, too, the best mixture lies between one-twelfth and one-fourteenth of gas ; with less and more gas the result becomes worse and worse. Glasgow and Oldham gases seem to be very nearly equal in value per cubic foot for the production of power, as the pressure produced from one cubic inch in the best mixture of each is very similar. The average pressures during 0·2 second from complete explosion are exceedingly close, Glasgow gas mixture containing one-twelfth gas giving 666 lbs. pressure per cubic inch of gas, and Oldham gas for the same mixture and the same quantity giving 630 lbs.: Glasgow gas

one-fourteenth mixture 665 lbs. pressure, Oldham gas 640 lbs. The hydrogen experiments give as follows :

Proportion of hydrogen gas in mixture .	$\frac{1}{7}$	$\frac{1}{5}$	$\frac{2}{7}$.
Pressure produced upon pistons by one cubic inch hydrogen	287,	340,	280.
Pressure remaining upon pistons 0·2 sec. after complete explosion per sq. inch .	35,	39,	40.
Pressure per piston	245,	195,	140.
Mean pressure upon piston . . .	266,	267,	210.

The best mixture with one cubic inch of hydrogen only gives a pressure of 267 lbs. available for 0·2 second, so that its capacity for producing power, compared with Glasgow and Oldham gas, is as 267 is to 665 and 640 respectively. Assuming the actual heat capacity of the mixtures to be equal, then to produce equal power with Glasgow gas nearly two-and-a-half times its volume of hydrogen is required. The idea is very prevalent among inventors that if pure hydrogen and air could be used, greater power and economy would be obtained; these experiments prove the fallacy of the notion. Hydrogen is a poor gas to use alone in the cylinder of a gas engine; it is useful in conferring inflammability upon dilute mixtures of other gases, but when present in large quantity in coal gas it diminishes its value per cubic foot for power.

Pressures produced if no Loss or Suppression of Heat Existed

From the fact already mentioned in the last chapter, that the theoretical temperatures of combustion are never attained in reality, it will naturally be expected that the pressures produced by explosions in closed vessels will also fall short of theory. This is found to be the case. It has been observed by every experimenter upon the subject, beginning with Hirn in 1861, who determined the pressures produced by the explosion of coal gas and air, and hydrogen and air. He used two explosion vessels of 3 and 36 litres capacity; they were copper cylinders with diameters equal to their length. He used a Bourdon spring manometer to register the pressure. He states that :

(1) With 10 per cent. hydrogen introduced the results were: according to experiment, 3·25 atmospheres ; according to calculation, 5·8 atmospheres.

(2) With 20 per cent. of hydrogen, the results were : according to experiment, 7 atmospheres, which is very much below the calculation.

(3) With 10 per cent. of lighting gas introduced the results were : according to experiment, 5 atmospheres, *i.e.* much more than with the introduction of an equal volume of pure hydrogen.

He notices especially the low pressure produced by hydrogen as compared with lighting gases, but observes truly that this should not excite surprise—although the heat value of hydrogen is great, yet it is so when compared with equal weights of other substances—and that coal gas being four or five times as heavy as hydrogen, quantity is balanced against quality; therefore volume for volume it gives out more heat.

He considers that there is no difficulty in explaining the very considerable difference found between calculation and experiment, as the metal sides are at so low a temperature compared with the explosion, that the heat is rapidly conducted away, and the attainment of the highest temperature is impossible. Bunsen, in his experiments, observed the same difference, and so later did Mallard and Le Chatelier. The author's experiments fully confirm the accuracy of those observers. In no case, whether with weak or strong mixtures of coal gas and air, or hydrogen and air, is the pressure produced which should follow the complete evolution of heat.

Thus, with hydrogen mixtures (*Clerk's experiments*):

	Per sq. in.		
1 vol. H, 6 vols. air, gives by experiment .	41	lbs.	above atmosphere.
The calculated pressure is	88·3	,,	,,
1 vol. H, 4 vols. air, experiment gives .	68	,,	,,
Calculated pressure is	124	,,	,,
2 vols. H, 5 vols. air, experiment gives . .	80	,,	,,
Calculated pressure is	176	,,	,,

Without exception the actual pressure falls far short of the calculated pressure; in some manner the heat is suppressed or lost. That the difference cannot altogether be accounted for by loss of heat is easily proved; the fall of pressure is so slow from the maximum that it is impossible that any considerable proportion of heat can be lost in the short time of explosion. If so large a proportion were lost on the rising curve, it could not fail to show upon the falling curve; it would fall, in fact, as quickly as it rose. Again, the increase of pressure would be less in a small than in a large vessel, as the small vessel exposes the larger surface proportionally to the gas present. It is found that this is not so. Bunsen used a vessel of a few cubic centimetres capacity, and got with carbonic oxide and oxygen true explosive mixture 10·2 atmospheres maximum pressure; Berthelot with a vessel 4000 cubic centimetres capacity got 10·1 atmospheres; with hydrogen true explosive mixture, Bunsen 9·5 atmospheres, Berthelot, 9·9 atmospheres. All the difference, therefore, cannot be accounted for by loss before complete explosion.

Mixtures of air and coal gas give similar results.

The following are the observed and calculated pressures for Oldham coal gas (*Clerk's experiments*) :

	Per sq. in.		
1 vol. gas, 14 vols. air, experiment gives	40 lbs.	above	atmosphere.
Calculated pressure is	89·5	,,	,,
1 vol. gas, 13 vols. air, experiment gives	51·5	,,	,,
Calculated pressure is	96	,,	,,
1 vol. gas, 12 vols. air, experiment gives	60	,,	,,
Calculated pressure is	103	,,	,,
1 vol. gas, 11 vols. air, experiment gives	61	,,	,,
Calculated pressure is	112	,,	,,
1 vol. gas, 9 vols. air, experiment gives	78	,,	,,
Calculated pressure is	134	,,	,,
1 vol. gas, 7 vols. air, experiment gives	87	,,	,,
Calculated pressure is	168	,,	,,
1 vol. gas, 6 vols. air, experiment gives	90	,,	,,
Calculated pressure is	192	,,	,,

The results with Glasgow gas are so similar that it is unnecessary to give a table; in no case does the maximum pressure account for much more than one-half of the total heat present.

It is to be noted that these calculations assume the specific heats determined by Regnault to be true for high temperatures. Regnault's numbers, as given at p. 118, have been used for the calculations.

As all of the deficit cannot have disappeared previous to complete explosion, it follows that the gases are still burning on the falling curve or that the specific heats have greatly increased. That is, the falling curve does not truly represent the rate of cooling of air heated to the maximum temperature, because heat is being continually added by the continued combustion of the mixture, or varying specific heat produces an effect similar to heat evolution by combustion.

It may, however, be taken as completely proved by the complete accord of all physicists who have experimented on the subject, that for some reason nearly one-half of the heat present as inflammable gas in any explosive mixture, true or dilute, is kept back and prevented from causing the increase of pressure to be expected from it on the assumption of constant specific heat. Although differences of opinion exist on the cause, all are agreed on the fact; they also agree in considering that inflammation is complete when the highest pressure is attained.

Temperatures of Explosion

With a mass of any perfect gas confined in a closed vessel the absolute temperatures and pressures are always proportional; double temperature means double pressure. Temperatures T, t (absolute), pressures corresponding P, p; then $\frac{T}{t}=\frac{P}{p}$ (Charles's law). If explo-

sive mixtures behaved as perfect gases, the pressure before explosion and temperature being known, the pressure of explosion at once gives the corresponding temperature. It has been shown at page 112 that explosive mixtures do not fulfil this condition, but change in volume from chemical causes quite apart from physical ones. It follows, therefore, that these changes must be known before the temperature of the explosion can be calculated from the pressure. In the cases of hydrogen and carbonic oxide true explosive mixtures with oxygen, a contraction of volume is the result of combination. It comes to the same thing as if a portion of the perfect gas in the closed vessel was lost during heating; the temperature, then, could not be known at the higher pressure unless the volume lost is also known.

Suppose one-third of the volume to disappear upon cooling to the original temperature the pressure would be reduced to two-thirds of the original pressure, and this fraction of the original pressure must be taken as $p_1 = 10$. As both steam and carbonic acid at temperatures high enough to make them perfectly gaseous occupy two-thirds of the volume of their free constituents, it follows that p_1 must be taken as $\frac{2}{3}p$, wherever the temperatures are such that combination is complete. But here another difficulty occurs. Bunsen found that hydrogen and oxygen in true explosive mixtures gave an explosion pressure of 9·5 atmospheres. The calculated pressure for complete combustion and allowing for chemical contraction is 21·3 atmospheres. It is evident enough that complete combustion has not occurred, but it is difficult to say what fraction remains uncombined. Yet unless the fraction in combination be known the contraction cannot be known, and therefore the temperature corresponding to the pressure cannot be known.

Berthelot has pointed out that in a case of this kind the true temperature cannot be calculated, but it may be shown to lie between two extreme assumptions, both of which are erroneous:

(1) Temperature calculated on assumption of no contraction.

(2) Temperature calculated on assumption of the complete contraction.

Let the two temperatures be (1) T^1 and (2) T.

	T^1	T
2 vols. H, 1 vol. O, explosion pressure (absolute) 9·9 atmospheres.	2449° C.	3809° C.
2 vols. CO, 1 vol. O, explosion pressure (absolute) 10·8 atmospheres	2612° C.	4140° C.

The lower temperature could only be true if no combination whatever had occurred, which is impossible, as then no heat at all could be evolved; the higher temperature could only be true if complete

combination, and therefore complete contraction, occurred The truth is somewhere between these numbers.

When the explosive mixture is dilute, the limits of possible error are narrower, because the possible proportion of contraction is less ; with hydrogen and air mixture in proportion for complete combination, 2 volumes of hydrogen require 5 volumes of air. The greatest possible contraction of the 7 volumes is therefore 1 volume. If all the hydrogen burned to steam, the 7 volumes contract to 6 volumes. With more dilute mixtures the proportion diminishes.

With a mixture containing $\frac{1}{5}$ of its volume hydrogen, 10 volumes can only suffer contraction to 9 volumes. With $\frac{1}{7}$ volume hydrogen, 14 volumes can contract to 13 volumes.

The limits of maximum temperatures for those mixtures are as follows (*Clerk*) :

	T^1	T
1 vol. H, 6 vols. air, explosion pressure (absolute), 55·7 lbs. per sq. in.	826° C.	909° C.
1 vol. H, 4 vols. air, explosion pressure (absolute), 82·9 lbs. per sq. in.	1358° C.	1539° C.
2 vols. H, 5 vols. air, explosion pressure (absolute), 94·7 lbs. per sq. in.	1615° C.	1929° C.

The possible error is here much less than with true explosive mixtures ; coal gas is of such a composition that some of its constituents expand upon decomposition previous to burning, and so to some extent balance the contraction produced by the burning of the others. The possible error is therefore still further reduced. The composition of Manchester coal gas as determined by Bunsen and Roscoe is as below. The oxygen required for the complete combustion of each constituent is also given, and the volumes of products formed.

Analysis of Manchester Coal Gas. (*Bunsen and Roscoe*)

—	—	Amount required for complete combustion	Products
	vols.	vols. O	vols.
Hydrogen, H.	45·58	22·79	45·58, H_2O
Marsh gas, CH_4	34·9	69·8	104·7, CO_2 and H_2O
Carbonic oxide, CO	6·64	3·32	6·64, CO_2
Ethylene, C_2H_4	4·08	12·24	16·32, CO_2 and H_2O
Tetrylene, C_4H_8	2·38	14·28	19·04, CO_2 and H_2O
Sulphuretted hydrogen, H_2S	0·29	0·43	0·58, H_2O and SO_2
Nitrogen, N.	2·46	—	2·46
Carbonic acid, CO_2	3·67	—	3·67
Total	100·00	122·86 O	198·99, CO_2, H_2O & SO_2

When burned in oxygen 100 volumes of this sample of gas require 122·86 volumes of oxygen, total mixture 222·86 volumes; the products of the combustion measure 198·99 volumes. Calculating to percentage, 100 volumes of the mixture will contract to 89·4 volumes of the products. As 100 volumes of the mixture will contain 55·1 volumes of oxygen, it follows that if air be used, four times that volume of nitrogen will be associated with it, that is, $55{\cdot}1 \times 4 = 220{\cdot}4$.

The strongest possible explosive mixture of this coal gas with air containing 100 volumes of the true explosive mixture will be 320·4 volumes, and it will contract upon complete combustion to 309·8 volumes.

One volume of this gas requires 6·14 volumes air for complete combustion, and 100 volumes of the mixture contract to 96·6 volumes of products and diluent, a contraction of 3·4 per cent. Dilution still further diminishes the change; thus a mixture, 1 volume gas 13·28 volumes air, will have only half that contraction, or 1·7 per cent.

From these figures it is evident that the limits of possible error in calculating temperature from pressure of explosion does not exceed, in the worst case, with coal gas and air 3·4 per cent., and in weaker mixtures half that number. The fact that the whole heat is not evolved at the explosion pressure, and that therefore the whole contraction does not occur then, further reduces the error. It is then nearly correct to calculate temperature from pressure without deduction for contraction. This has been done for Glasgow gas and for the Oldham gas experiments by the author.

EXPLOSION IN A CLOSED VESSEL. (*Clerk*)

Mixtures of air and Glasgow coal gas

Temp. before explosion . . . . 18° C.
Pressure before explosion . . . . atmos. 14·7 lbs.

Mixture		Max. press. above atmos. in pounds per sq. in.	Temp. of explosion calculated from observed pressure
Gas	Air		
1 vol.	13 vols.	52	1047° C.
1 vol.	11 vols.	63	1265° C.
1 vol.	9 vols.	69	1384° C.
1 vol.	7 vols.	89	1780° C.
1 vol.	5 vols.	96	1918° C.

Mixtures of air and Oldham coal gas

Temp. before explosion . . . . 17° C.

Mixture		Max. press. above atmos. in pounds per sq. in.	Temp. of explosion calculated from observed pressure	Theoretical temp. of explosion if all heat were evolved
Gas	Air			
1 vol.	14 vols.	40	806° C.	1786° C.
1 vol.	13 vols.	51·5	1033° C.	1912° C.
1 vol.	12 vols.	60	1202° C.	2058° C.
1 vol.	11 vols.	61	1220° C.	2228° C.
1 vol.	9 vols.	78	1557° C.	2670° C.
1 vol.	7 vols.	87	1733° C.	3334° C.
1 vol.	6 vols.	90	1792° C.	3808° C.
1 vol.	5 vols.	91	1812° C.	
1 vol.	4 vols.	80	1595° C.	

Those temperatures calculated from maximum pressure, although not quite true are very nearly so, whatever be the theory adopted to explain the great deficit of pressure. It does not follow, however, that they are the highest temperatures existing at the moment of explosion; they are merely averages. The existence of such an intensely heated mass of gas in a cold cylinder causes intense currents, so that the portion in close contact with the cold walls will be colder than that existing at the centre. There will be a hot nucleus of considerably higher temperature than that outside, but whatever that temperature may be, the increase of pressure gives a true average. It may be taken, then, that coal gas mixtures with air give upon explosion temperatures ranging from 800° C. to nearly 2000° C., depending on the dilution of the mixture. The more dilute the mixture the lower the maximum temperature; increase of gas increases maximum temperature at the same time as it increases inflammability.

The author has made explosion experiments in the same vessel with mixtures previously compressed, and finds that the pressures produced with any given mixture are proportional to the pressure before ignition, that is, with a mixture of constant composition, double the pressure before explosion, keeping temperature constant at 18° C., doubles the pressure of explosion.

Efficiency of Gas in Explosive Mixtures

Rankine defines available heat as follows:

'The available heat of combustion of one pound of a given sort of fuel is that part of the total heat of combustion which is communicated to the body to heat which the fuel is burned; and the efficiency of a given furnace, for a given sort of fuel, is the proportion which the available heat bears to the total heat.'

The gas engine contains furnace and motor cylinder in one; nevertheless the efficiency of the working fluid is quite as distinct from the furnace efficiency as in the steam engine. Rankine's definition is quite true for the gas engine.

The fuel being gas, the working fluid consists of air and its fuel and their combinations; the available heat is that part of the heat of combustion which serves to raise the temperature of the working fluid; the part which flows into it to make up for loss to the cold cylinder walls cannot be considered available. To be truly available it must either increase temperature, or keep it from falling by expansion. The heat flowing through the cylinder walls is a furnace loss, incident to the explosion method of heating.

The experiments upon explosion in a closed vessel provide data for determining the furnace efficiency as distinguished from that of the working fluid. The proportion of heat flowing from an explosion to the walls in unit time will depend upon the surface of the walls for any given volume. The smaller the cooling surface in proportion to volume of heated gases, the slower will be the rate of cooling. Therefore to be applicable to any engine, the explosion vessel in which the experiments are made should have the same capacity and surface as the explosion space of the engine.

The author's experiments are therefore only strictly applicable to engines with cylinders similar to his explosion vessel. Within certain limits, however, the error introduced by applying them to other engines is not large.

Assuming the stroke of a gas engine (after explosion) to take 0·2 second, this may be taken as the time during which the pressure of explosion must last if it is to be utilised by the engine. In a closed vessel the pressure falls considerably in 0·2 second: the average pressure may be taken as nearly indicating the available pressure during that time. The heat necessary to produce that pressure is the available heat; and its proportion to the total heat which the gas present in the mixture can evolve is the efficiency of the gas in that explosive mixture.

With Oldham gas the best mixture is (table, p. 133) 1 volume gas 12 volumes air; the average pressure during the first fifth of a second is 51 lbs. per sq. in. above atmosphere. If all the heat present heated the air, the pressure should be 103 lbs. effective, so that the efficiency of the heating method is $\frac{51}{103} = 0{\cdot}49$.

The strongest mixture which still contains oxygen in excess is 1 volume gas 7 volumes air, the average available pressure is 67 lbs. per sq. in. (all heat evolved would give 168 lbs.), the efficiency is $\frac{67}{168} = 0{\cdot}40$ nearly.

Calculated in this way the efficiency values for Oldham gas mixtures are:

Prop. of Oldham gas in mixture	$\frac{1}{15}$,	$\frac{1}{14}$,	$\frac{1}{13}$,	$\frac{1}{12}$,	$\frac{1}{10}$,	$\frac{1}{8}$,	$\frac{1}{7}$
Heating efficiency . . .	0·40,	0·48,	0·50,	0·43,	0·46,	0·40,	0·37.

The furnace efficiency plainly diminishes with increased richness of the mixture in gas. These calculations, however, assume constant specific heat and completed combustion, and therefore include more than furnace loss.

Time of Explosion in Closed Vessels

The rates of the propagation of flame in explosive mixtures given in tables, pages 113 and 115, are true only where the inflamed portion is free to expand without projecting itself into the unignited portion. They are the rates proper for constant pressure.

Where the volume is constant, in a closed vessel, the part first inflamed instantly expands and so projects the flame surface into the mass, compressing what remains into smaller space.

To the rate of inflammation at constant pressure is added the rate of projection of the flame into the mass by its expansion and also the increased rate of propagation in the unignited portion by the heating due to its compression by the portion first inflamed.

It follows that the rate continually increases, as the inflammation proceeds until it fills the vessel.

This is evident from all the explosion curves. The pressure rises slowly at first, then with ever-increasing rate till the explosion is complete; thus the explosion curve for hydrogen mixture with air $\left(\frac{2}{7}H\right)$ shows an increase of 17 lbs. in the first 0·005 second, the maximum pressure of 80 lbs. being attained in the next 0·005 second. With the weaker mixtures the same thing occurs, rise of pressure, slow at first, then more rapid, and in some cases becoming slow again before maximum pressure. The time taken to get maximum pressure varies much with the circumstances attending the beginning of the ignition. If a considerable mass be ignited at once, by a long and powerful spark, or by a large flame, the ignition of the weakest mixture may be made almost indefinitely rapid. Something very like Berthelot's explosive wave may result. This is due to the great mechanical disturbance caused by the rapid expansion of the portion first ignited; the smaller that portion is, the more gently does the flame spread. A small separate chamber connected with the main vessel, if filled with explosive mixture and ignited, will project a rush of flame into the main vessel and cause almost instantaneous ignition. The shape

of the vessel, too, has a great effect upon the rate. Where it is cylindrical and large in diameter proportional to its axial length, ignition is extremely rapid, the flame is confined at starting, and is rapidly deflected by the cylinder ends, and so shoots through the whole mass.

By so arranging the explosion space of a gas engine that some mechanical disturbance is permitted, it is easy to get any required rate of ignition even with the weakest mixtures.

The maximum pressure is not increased by rapid ignition.

Starting the ignition from a small spark, the time taken to ignite increases with the volume of the vessel.

Berthelot has experimented upon this point with explosion vessels of three capacities, 300 cubic centimetres, 1500 cubic centimetres, and 4000 cubic centimetres. He finds time of explosion (he also takes

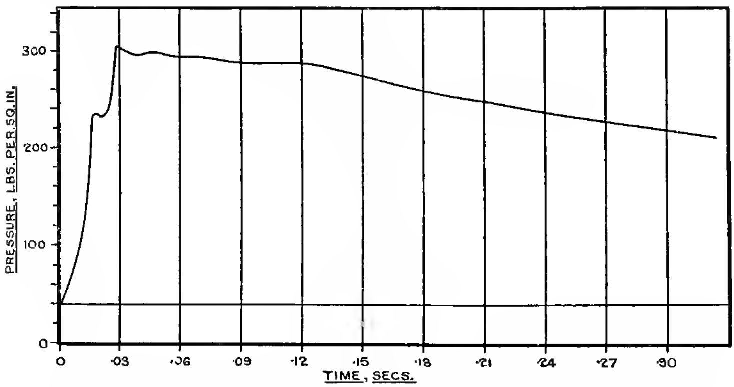


FIG. 38.—Explosion of Oldham Coal Gas and Air Mixture in Closed Vessel, with previous compression of 40 lbs. per sq. in. above atmos.

maximum pressure to indicate complete explosion) of mixture 2 vols. H, 1 vol. O, and 2 vols. N, in 300 cubic centimetre vessel, 0·0026 second; and in 4000 cubic centimetre vessel, 0·0068 second.

With mixture of carbonic oxide and oxygen, 2 vols. CO, 1 vol. O, smaller vessel, 0·0128 second; larger vessel, 0·0155 second. Mixtures with air were much slower. The conclusion, then, is obvious, that in large engines the time of explosion will be longer than in small ones.

Later Experiments. Explosions in Closed Vessels

The experiments on gaseous explosion in a closed vessel were began by the author in 1883, and in part communicated to the late Professor

Fleeming Jenkin for the purpose of a lecture delivered by him to the Institution of Civil Engineers on February 21, 1884. The hydrogen and Glasgow coal gas curves were published in that lecture. Other experiments were made in 1885, and all the foregoing results were published in a paper read by the author on March 9, 1886.

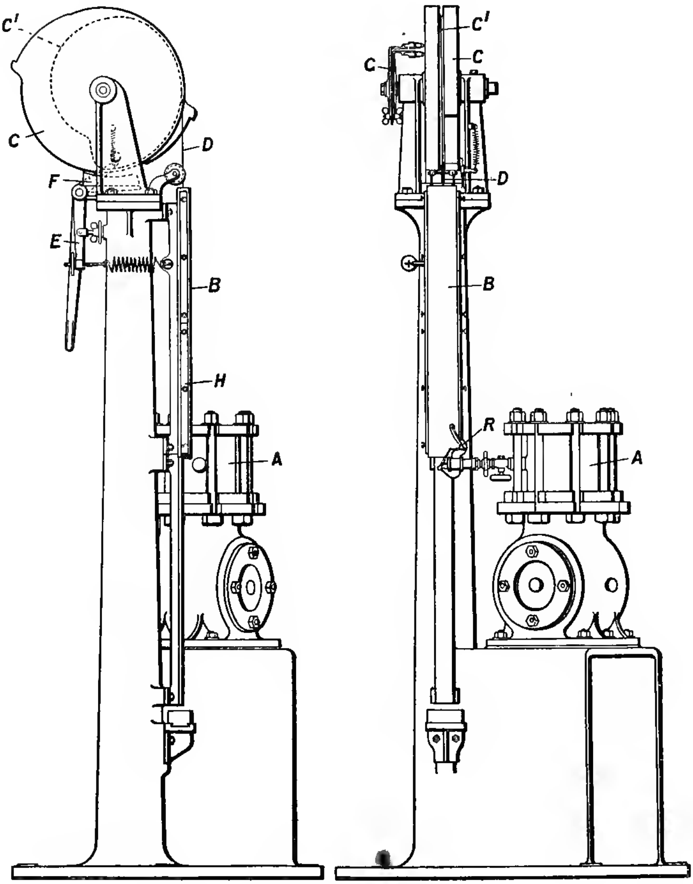


FIG. 39.—Clerk Explosion Apparatus of 1900

Mixtures compressed before ignition were also tested in the same vessel. Fig. 38 shows one of the curves published in a paper by the author read to the Society of Chemical Industry on January 29, 1886. Further experiments were made in 1900 with the author's new apparatus.

Clerk's Later Apparatus, 1900

The apparatus is shown at fig. 39 to consist of a closed cylindrical vessel 7 ins. diameter, 7 ins. long, internal measurement, 269 cubic ins. capacity, bored, turned, and supplied with strongly bolted covers as in the earlier device. A Richards indicator was mounted upon the upper cover, as shown, and provision was made for firing electrically. The rotating drum used in the earlier apparatus was dispensed with, and a long falling platform or slide was adopted. The object was to obtain a diagram on a long indicator card, so that no transfer from a drum to a tracing was required. The card was over 30 ins. long. The long slide carrying the paper was fitted accurately

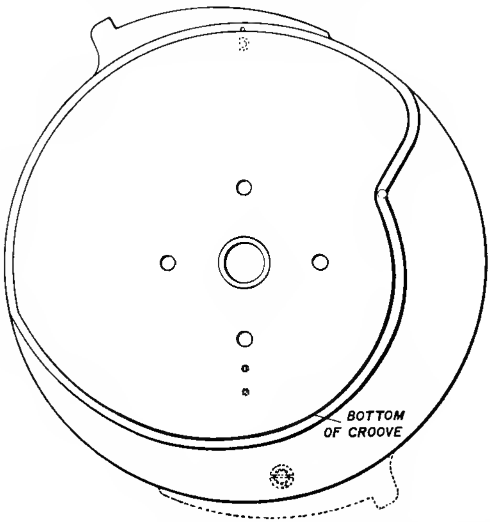
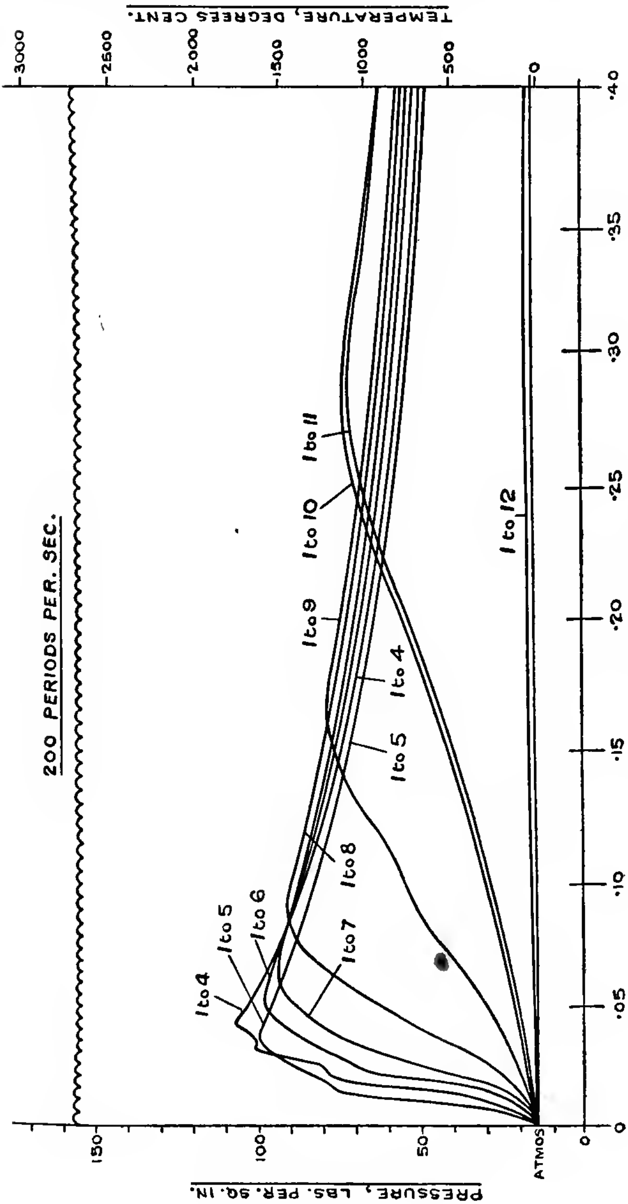


Fig. 40.—Outline of Cam Groove in Clerk Explosion Apparatus

on V slides, and it was suspended by a pianoforte wire from a cam groove in a flywheel placed above it. The cam groove resembled the fusee of a watch, and the curve was so constructed that when a lever was operated the flywheel was let go and the weight of the slide forced it to rotate. For the first 3 ins. of the fall the motion of the slide accelerated, but then the pianoforte wire arrived at a part of the cam surface of smaller diameter than at first, and the diameter progressively diminished, so that as the flywheel accelerated the diameter diminished. In this way, after 3 ins. fall, the motion of the slide remained uniform until at the end of its travel a brake applied to the flywheel stopped its motion. This chronograph resembles the falling

bar chronograph used to determine the velocity of a bullet, but the fall of the slide is controlled by the flywheel instead of being free. The



Analysis of Coal Gas

Hydrogen	45·2
Marsh gas	29·9
Unsaturated hydrocarbons	5·1
Carbonic oxide	10·3
Oxygen	0·5
Nitrogen	10·0
	100·1

FIG. 41.—Explosion in a Closed Vessel. London Gas, 1900. (*Clerk*)

time of fall is determined and checked by means of a tuning-fork operated electrically and actuating an electrical relay which traces a curve on the falling paper of 200 periods per second. Once the apparatus is calibrated by this tuning-fork, it can be used for a great number of successive experiments. The operations of charging the explosion vessel, measuring the gas, and so forth are the same as were used in the earlier device.

On fig. 39, A is the explosion vessel with its indicator, which in this case had a metallic pencil, B is the falling slide, C the flywheel with its cam groove C_1, shown on a larger scale at fig. 40. D is the pianoforte wire, E the trigger lever for letting go the slide, and F the wedge brake for stopping the flywheel at the end of its movement; G is an electric contact arrangement, arranged to pass the electric spark through the explosion vessel at any period of the movement of the flywheel; H is the clamping device to stretch the metallic paper and hold it on the falling slide; R is the Richards indicator in position.

This apparatus is very useful for performing a great number of experiments, as it requires no motive power to keep a drum in rotation, and the parts are so solidly constructed that derangement due to change in friction is easily avoided. Experiments at atmospheric pressure with London coal gas were made with this apparatus, and the diagrams of one set are shown at fig. 41.

The analysis of the London coal gas used is given below the diagrams.

These explosions with London coal gas give the following maximum pressures, temperatures, and times of explosion:

EXPLOSION IN A CLOSED VESSEL. (*Clerk* 1900)

Mixtures of air with London Coal Gas

Temp. before explosion . . . . 16° C.
Pressure before explosion . . . 14·8 lbs. per sq. in.

Mixture		Max. Pressure above atmosphere in lbs. per sq. in.	Temp. of Explosion from observed pressure	Time of Explosion
Gas	Air			
1 vol.	12 vols.	4	—	—
1 vol.	11 vols.	58	1150° C.	0·290 sec.
1 vol.	10 vols.	60	1155° C.	0·305 sec.
1 vol.	9 vols.	65	1270° C.	0·155 sec.
1 vol.	8 vols.	77	1475° C.	0·087 sec.
1 vol.	7 vols.	80	1565° C.	0·067 sec.
1 vol.	6 vols.	85	1660° C.	0·055 sec.
1 vol.	5 vols.	87	1710° C.	0·042 sec.
1 vol.	4 vols.	93	1830° C.	0·045 sec.

Calculating the best mixtures as before, it is found that 1 gas to 11 of air is best both for maximum pressure and duration for $\frac{1}{5}$ second.

Massachusetts Institute of Technology Experiments

In 1898 an apparatus somewhat similar to the author's was constructed at the Massachusetts Institute of Technology at Boston, and

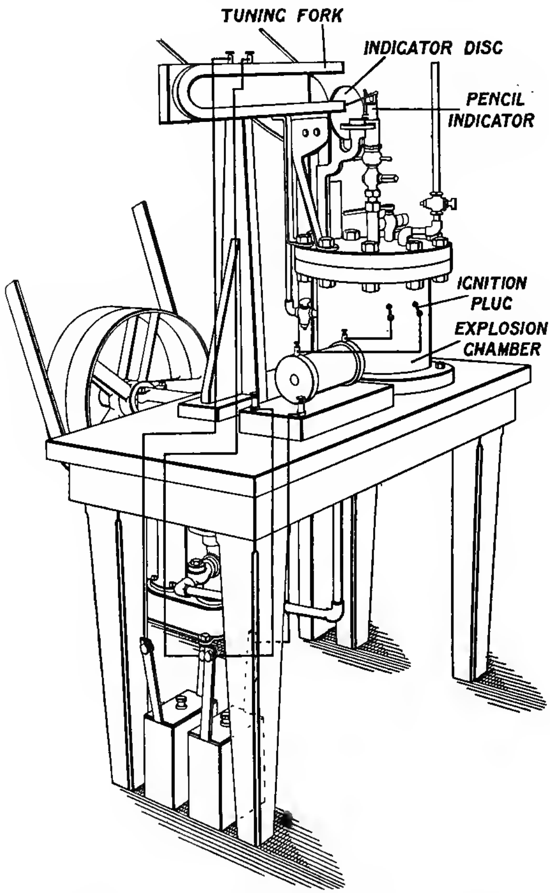
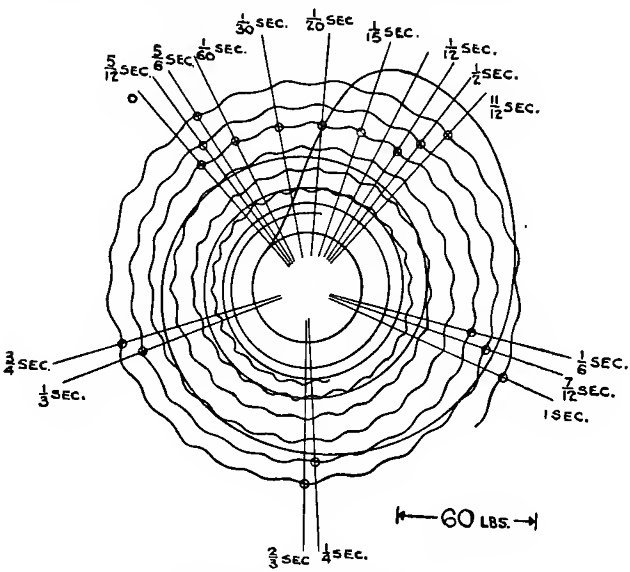


Fig. 42.—Massachusetts Institute of Technology—Explosion Apparatus

many interesting experiments have been made with it. It consists of a cast-iron cylinder of 310 cubic ins. capacity, so that it closely corresponds to the Clerk apparatus of 317 ins. capacity. To introduce the mixture the explosion vessel is exhausted by a pump and

scavenged by admitting fresh air; this operation is repeated sufficiently to clear the cylinder from the products of the previous explosion. The cylinder is then exhausted down to a carefully measured pressure and coal gas is admitted to raise the pressure to atmosphere again. By this device any desired proportion of gas may be mixed with air within the cylinder. In the early Clerk experiments the record of explosion and cooling was taken upon a rotating drum; in the Boston experiments a power-driven disc was adopted, and the line was traced by the indicator on the face of this disc. The disc at the same time receives a time tracing from a pointer attached to one arm of a tuning-



Mixture: 1 vol. Boston gas, 9 vols. air

FIG. 43.—Record from Explosion Apparatus

Massachusetts Institute of Technology, Boston

fork, which is kept in movement electrically. The mixtures are fired electrically.

Fig. 42 shows a general view of the apparatus, and fig. 43 shows the record as taken on the disc. These diagrams are taken on the base of a spiral line, and, although mechanically convenient, this leads to increased work in reducing the observations.

Fig. 44 shows a set of these diagrams developed upon a straight line base, so that they directly compare with the Clerk diagrams already discussed.

The analysis of the Boston coal gas used was as follows:

ANALYSIS OF BOSTON COAL GAS

Massachusetts Institute of Technology

	Per cent.
Carbonic oxide, CO	25·3
Illuminants	12·0
Carbonic acid, CO_2	1·9
Marsh gas, CH_4	28·9
Nitrogen, N	3·0
Hydrogen, H	27·9
Oxygen, O	1
	100·0

The percentage of gas present is marked in the corresponding curve at fig. 44, and proportions by parts are also marked.

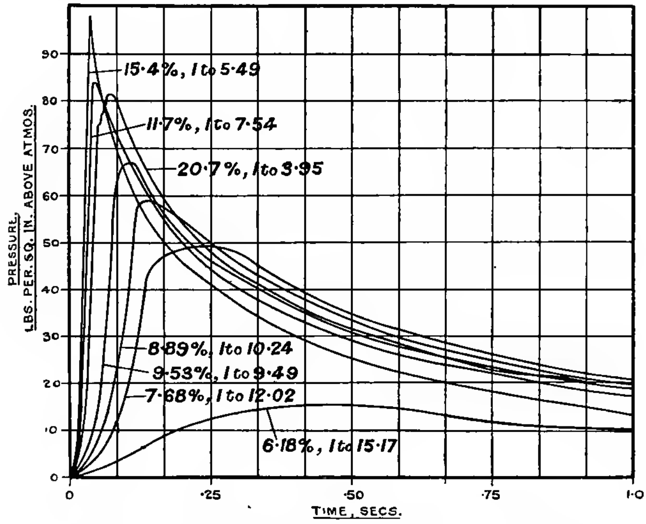


FIG. 44.—Explosion of Gaseous Mixtures in a Closed Vessel. Boston Experiments

The Clerk method of determining the efficiency was used, and the same period of $\frac{1}{5}$ second from explosion and maximum pressure was chosen, so as to make the observations comparable with those of Clerk. As the vessel was of the same capacity, the comparison is very close. Two periods of $\frac{1}{5}$ second were examined—the first timing from the moment of firing, and the second from the moment of maximum pressure.

The following table shows the main results, comprising the examn ation of the cooling curve to $\frac{1}{5}$ second from maximum pressure.

EXPLOSION IN A CLOSED VESSEL. (*Boston Experiments*)

Mixtures of Air and Boston Coal Gas

Temperature and Pressure before explosion Atmospheric

Mixture Gas: Air	Max. pres. lbs. per sq. in.	Time of Explosion	Area, sq. ins.	Mean pres. lbs. per sq. in.	Mean pres. / ϕ gas ratio	Final pressure	Final pressure / ϕ gas ratio
1	2	3	4	5	6	7	8
		Sec					
1— 3	45	0·49	—	43	172	40	160
1— 4	86	0·08	1·88	62	310	46	230
1— 5	96	0·05	1·93	64	384	44	264
1— 6	88	0·05	1·93	64	448	46	322
1— 7	86	0·06	1·93	64	512	48	384
1— 8	87	0·06	1·83	61	549	46	414
1— 9	77	0·08	1·86	62	620	46	460
1—10	71	0·11	1·69	56	616	45	495
1—11	68	0·14	1·66	55	660	43	516
1—12	39	0·33	0·98	33	429	30	390
1—13	32	0·42	0·79	26	364	24	336
1—14	9	0·42	0·24	8	120	8	120

The only column which requires a word of explanation is Column 4. Area in square inches signifies the area under the cooling curve from maximum pressure to $\frac{1}{5}$ second after, and the mean pressure in Column 5 is taken from this area as determined by planimeter. This is a more accurate method than taking the mean of the maximum pressure and pressure $\frac{1}{5}$ second after maximum, as was done by the author. The final pressure after the $\frac{1}{5}$ second from maximum is given at Column 7. The difference between the methods is not great, as will be seen by taking out the means of the second and seventh columns. Column 6 gives the numbers obtained by dividing the mean pressure on Column 5 by the gas ratio. This is what was done by Clerk, as already described. The numbers in Column 6, then, give not only the relative values of the different mixtures for maximum pressure and resistance to cooling in a closed vessel, but they enable Boston gas to be compared with Glasgow, Oldham, and London gases of the author's experiments.

The Boston experiments show the best mixture to be 1 gas and 11 air, or $\frac{1}{12}$ of gas, as shown by the author's experiments to be true for both Oldham and Glasgow. The comparable numbers for $\frac{1}{12}$ for Oldham and Boston are respectively 630 and 660, so that the gases vary but little in power-producing value.

The Boston experiments closely correspond in other results with the author's, as will be discussed later. Interesting experiments have also been made at the same Institution with mixtures of air and petrol vapour, using the same apparatus and the Clerk method of comparison. The two tables on p. 152 are similar to that just described.

EXPLOSION IN A CLOSED VESSEL. (*Boston Experiments*)

Mixtures of Air and Petrol Vapour.

Petrol sp. gr. 0·648 at 86° F.

Percentage of petrol vapour in mixture	Time of explosion. Seconds	Max. press. in lbs. per sq. in. above atmosphere	0·2 sec. after maximum pressure			
			Area, sq. in.	Mean press. lbs. per sq. in.	Mean press. ÷ vapour ratio	Final pressure
1·51	0·083	70	1·48	49·4	3260	34
1·64	0·100	73	1·53	51·0	3110	36
1·79	0·090	71	1·43	47·7	2670	33
1·96	0 083	76	1·55	51·7	2634	35
2·17	0·058	70	1·45	48·4	2225	30
2·44	0·067	80	1·60	53·4	2190	36
2·56	0·075	84	1·69	56·4	2200	40
2·63	0·059	86	1·71	57·0	2164	38
2·78	0·083	78	1·62	54·0	1945	36
3·03	0·091	76	1·60	53·4	1760	38
3·23	0·083	77	1·62	54·0	1675	37
3·45	0·083	77	1·64	54·7	1587	37
3·85	0·075	66	1·50	50·0	1300	38
4·17	0·066	60	1·38	46·0	1104	35
4·76	0·066	56	1·32	44·0	925	33

Petrol sp. gr. 0·680 at 76° F.

Percentage of petrol vapour in mixture	Time of explosion. Seconds	Max. press. in lbs. per sq. in. above atmosphere	0·2 sec. after maximum pressure			
			Area, sq. in.	Mean press. lbs. per sq. in.	Mean press. ÷ vapour ratio	Final pressure
1·32	0·167	52	1·28	42·7	3240	33
1·41	0·117	62	1·42	47·3	3360	35
1·51	0·109	64	1·45	48·6	2950	35
1·64	0·182	51	1·25	41·7	2540	32
1·79	0·109	67	1·53	51·0	2855	36
1·96	0·091	73	1·53	51·0	2600	36
2·17	0·082	76	1·56	52·0	2391	37
2·44	0·060	85	1·63	54·3	2225	36
2·63	0·058	85	1·62	54·0	2052	36
2·78	0·058	84	1·64	54·7	1970	38
3·03	0·066	78	1·60	53·4	1760	37
3·23	0·067	83	1·70	56·7	1760	38
3·45	0·100	75	1·59	53·0	1536	38
3·85	0·117	62	1·42	47·3	1230	35
4·17	0·133	55	1·40	46·7	1121	38
4·76	0·210	35	1·02	34·0	714	32

These experiments show clearly that the best mixture of petrol and air with petrol of 0·648 sp. gr. is 1·51 per cent. of petrol vapour in the air mixture and with petrol of 0·680 sp. gr. 1·41 per cent. The mean pressure measuring the resistance to cooling and best pressure

is in the first case 49·4 lbs., and in the second 47·3 lbs. per sq. in.; while the best mean pressure with Boston gas is 55 lbs. per sq. in. According to this comparison a petrol engine using these petrol samples should give less power than a gas engine of the same volume swept by the cylinder.

As a rule it is found that petrol engines give rather higher mean pressures than gas engines. Some of the conditions in the petrol experiments appear to prevent a true comparison. Nevertheless the experiments are interesting and valuable, as but few explosion experiments have been made with petrol as yet.

Grover's Experiments

Interesting experiments were made by Mr. F. Grover in 1895 at the Yorkshire College of Science, Leeds. His explosion chamber was of 1 cub. ft. capacity, cylindrical, and internal dimensions about 8 ins. diameter, 34 ins. length. Mr. Grover states his object as follows:

'Experiments previously made upon gaseous mixtures have been directed towards the investigation of the actual pressures produced by the combustion of an inflammable gas, in the presence of oxygen or pure air only.' . . . 'The most complete practical contribution upon this subject has been afforded by the experiments of Mr. Dugald Clerk, which enabled him to estimate the most economical mixture to be used in a non-compression engine, but no account was taken of the effects of the products of combustion which are preserved in the cylinder of a gas engine, notwithstanding that early engines were constructed with a clearance volume of 60 per cent. To obtain some definite data upon this important subject the author (Mr. Grover) has carried out a series of experiments in the engineering laboratory of the Yorkshire College, Leeds.'

Mr. Grover states that products of combustion when mixed with fresh charge have been generally supposed to reduce the pressure produced by explosion, and therefore reduce the efficiency of the charge. Mr. Grover's experiments, however, lead him to the opposite conclusion. He says: 'The experiments carried out by the author show that the presence of the products of combustion in certain mixtures actually raise rather than diminish the maximum pressure obtained.'

This is a most important conclusion, if correct. But is it correct?

Mr. Grover's apparatus consisted of a thick cast-iron cylinder, flanged at both ends, as has been already stated, of 1 cubic foot capacity. The cylinder was bolted vertically to a column; the charge was ignited by passing an electric spark at the upper part of the vessel. Temperature before ignition was measured by a thermometer enclosed in a wrought-iron case containing mercury, inserted into the gas

chamber. The pressure changes were recorded by a Crosby indicator, the pencil of which was arranged to inscribe upon a continuously revolving drum, 8 ins. in diameter, driven by clockwork. The speed of the drum was checked by a vibrating spring adjusted to make four complete oscillations per second. The recording apparatus is shown diagrammatically at fig. 45.

In all the experiments Mr. Grover states that the volumes were measured by filling the cylinder with water and afterwards allowing the gas to enter as the water flowed out. The products of a previous combustion were retained to the desired extent in order to mix in the required proportions with the fresh charge. In some experiments the charge experimented upon consisted of gas and pure air only.

In making the pure air experiments, the explosion vessel was filled with water to expel the products, the water was then run out to draw into the vessel half the total volume of pure air required, then the coal

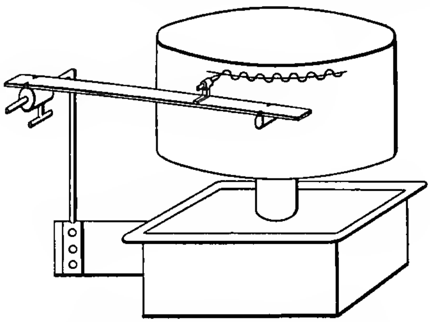


FIG. 45.—Grover Recording Apparatus

gas was admitted, and lastly the remaining half of the air required to complete the charge. Mr. Grover states that in all experiments, 'no appreciable time was allowed for the diffusion of the mixture, it having been fired immediately after taking the temperature.'

The coal gas used was taken from the service pipes of the Leeds gas-supply: the mean of three samples taken after the experiments were made gave the following analysis:

ANALYSIS OF LEEDS COAL GAS. (*Grover*)

Constituents	Volume per cent.
Marsh gas	35·2
Olefines	4·2
Hydrogen	52·9
Carbon monoxide	6·5
Nitrogen	0·1
Carbon dioxide and oxygen	1·1
	100·0

Mr. Grover, from air experiments, gave the results collected by the author in the following table :

EXPLOSION IN A CLOSED VESSEL

Pressure and Temperature before explosion. . . Atmospheric

Mixture		Maximum pressure above atmosphere in lbs. per sq. in.			
Gas	Air	Grover, 1895	Clerk, Oldham Gas	Clerk, London Gas	Massachusetts Inst. Boston Gas
		lbs.			
1	15	16	—	—	—
1	14	24	40	—	9
1	13	31	51·5	—	32
1	12	36	60	—	39
1	11	—	—	—	68
1	10	48	—	60	71
1	9	—	78	65	77
1	8	62	—	77	87
1	7	—	87	80	86
1	6	62	90	85	88

It will be noted that the explosion pressures produced in these experiments of Mr. Grover are much below those shown for similar mixtures by the experiments of Clerk and the Massachusetts Institute of Technology, which shall be called shortly the Boston experiments. To enable the comparison to be easily made the explosion pressures shown for similar mixtures of Oldham, London, and Boston coal gas are given in the last three columns. It will be observed that Grover's experiments agree with the Boston experiments ; for the mixtures containing $\frac{1}{14}$ and $\frac{1}{13}$ of gas the maximum pressures are very nearly the same, but the mixtures $\frac{1}{11}$, $\frac{1}{9}$, and $\frac{1}{7}$ give explosion pressures 30 per cent. lower. Compared with the Clerk Oldham gas experiments, mixtures $\frac{1}{15}$, $\frac{1}{14}$, and $\frac{1}{13}$ are 40 per cent. below, and mixtures $\frac{1}{11}$, $\frac{1}{9}$, and $\frac{1}{7}$ are 30 per cent. below, as in the Boston experiments.

The three sets of experiments are compared by the curves shown at fig. 46, where explosion pressures in pounds per square inch above atmosphere are shown on the vertical scale, and the percentage of gas in total volume of mixture is shown on the horizontal scale. The curves show considerable discrepancies. The *Clerk* curve for Oldham gas, it will be observed, smoothly passes through all the experimental points, and the *Grover* curve passes smoothly through all the points except the $\frac{1}{9}$ mixture; neglecting that point, the curve is a smooth one, resembling the Clerk curve, but at a lower pressure. The *Boston* curve is very irregular, and crosses both *Clerk* and *Grover* curves, but in its last six observations it agrees substantially with the Clerk curve. All these samples of coal gas have a lower calorific value of

nearly 600 B.Th.U. per cubic foot, so that they should show substantially the same maximum pressure for the same mixture. The Clerk curve for London gas is slightly below that for Oldham gas, the calorific value for London gas being a little lower.

Why are the results to some extent divergent? Mr. Grover's view is that his explosion pressures are lower than the author's because of his method of charging his cylinder by means of water, which leaves the whole of the interior surface in a moist condition. No doubt this explains some of the deficit; the water film would exercise some cooling effect, but this could not produce anything approaching 30 per cent. or 40 per cent. The author's experience convinces him that maximum pressure is but little checked in this way except in the case of the weakest mixtures. This water film explanation cannot account for

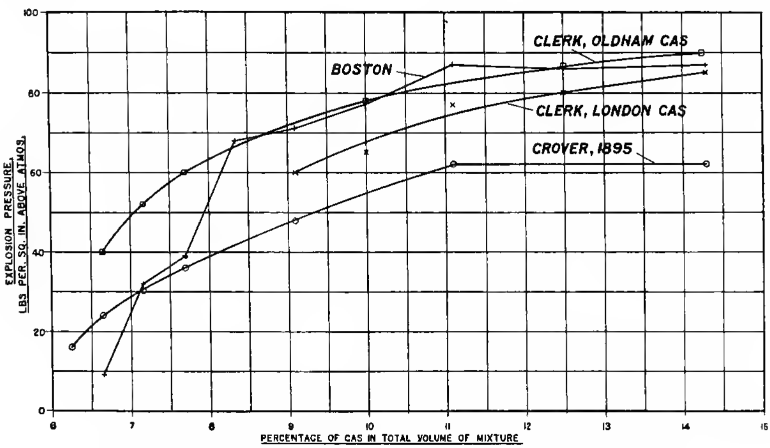


FIG. 46.—Comparison of Grover, Clerk, and Boston Experiments

the departure of the lower three Boston experiments from *Clerk's* curve. In the author's view such differences arise entirely from imperfect mixing of the gas and air contained in the explosion vessel. It was found by him in the course of the 1885 experiments that unless gas and air be well mixed low maximum pressures would be found. It was also found that weaker mixtures could be fired when the mixture was not uniform, provided some strong mixture was situated near the igniting point. In the author's early experiments he carefully experimented in order to determine this mixture point. In a paper 'On the Explosion of Homogeneous Gaseous Mixtures,' Inst.C.E., 1886, he states: 'To charge the explosion vessel, it is exhausted by an air pump, and a measured quantity of the inflammable gas admitted from a graduated glass measuring-vessel; air is

then let in, bringing the pressure up to the atmosphere again. The inflammable gas, being admitted while the vessel is almost vacuous, diffuses throughout the whole space, and the air entering afterwards, the contents are mixed thoroughly. To make perfectly sure that mixture is complete, the vessel is allowed to stand for at least half an hour before the explosion. To check the results obtained in this manner, a mixture of gas and air in a separate gasometer was made, and introduced into the explosion vessel by repeated exhaustion and filling ; the results were precisely similar, and the first method was used in most of the experiments, being more accurate and much safer.'

Clerk's London gas experiments were performed by the separate gasometer method, and here the effectiveness of mixing was undeniable. The London gas curves show the same higher pressures of explosion and give smooth curves.

The method of charging used in the Boston experiments, in which the gas added was measured in by reducing the pressure within the vessel and then letting gas flow in to fill up to atmosphere, would permit of very indifferent mixing, especially at the weak mixtures, where the pressure differences were small. Where this difference was greater better mixing would take place. The lower pressure obtained in weak mixtures the author considers to be entirely due to bad mixing. Again, in both *Boston* and *Grover* experiments weaker mixtures were found to be ignitable than was found by the author, and this also points to bad mixing.

Mr. Grover's experiments with exhaust gas admixtures were made as follows :

(1) Water was admitted to the cylinder to discharge part of the products of a previous combustion.

(2) Water was run out, so as to aspirate inwards half the volume of pure air required for the experiment.

(3) Air-cock was shut, and coal gas was drawn in by the same method.

(4) Water was entirely run out from the cylinder and the remaining air drawn in.

(5) The mixture of gas, air, and products of previous combustion were ignited at once by the electric spark and the time-pressure diagram taken.

Experiments made in this way showed that for equal dilutions made with air wholly and products of combustion and air, higher explosion pressures were recorded when products of combustion were present. That is, assume the explosion vessel to contain $\frac{1}{16}$, $\frac{1}{15}$, $\frac{1}{14}$, $\frac{1}{13}$, $\frac{1}{11}$ of its total volume of coal gas by measurement than when products of a previous combustion formed part of the $\frac{15}{16}$, $\frac{14}{15}$, $\frac{13}{14}$, $\frac{12}{13}$, or $\frac{10}{11}$ forming the other contents, higher explosion pressures were obtained than when

pure air only was used. The increase of pressure obtained, however, became less and less as the proportion of gas present increased, and when mixtures containing $\frac{1}{9}$ and $\frac{1}{7}$ of gas were tested no increase was found, but a diminution instead.

Mr. Grover's air and products of combustion mixtures gave results of which the most important are shown in the following table :

Explosion in a Closed Vessel. (*Grover* 1895)

Mixtures of Leeds gas, air, and products of combustion.
Pressure before explosion. . . Atmospheric.

Mixture					—	
Leeds Gas		Air	Products of combustion	Total	Max. pressure lbs. per sq. in. above atmosphere	—
Fraction	Per Cent.	Per Cent.	Per Cent.			
$\frac{1}{16}$	6·2	93·8	none	100	16	
,,	6·2	88·8	5	100	22	Maximum rise of pressure above that of pure mixture, 35 − 16 = 19 lbs. per sq. in.
,,	6·2	68·8	25	100	34	
,,	6·2	63·8	30	100	35	
$\frac{1}{15}$	6·6	93·4	none	100	24	Maximum rise of pressure above that obtained with pure mixtures, 36 − 24 = 12 lbs. per sq. in.
,,	6·6	88·4	5	100	24	
,,	6·6	68·4	25	100	28	
,,	6·6	63·4	30	100	36	
$\frac{1}{14}$	7·1	92·9	none	100	31	Maximum rise of pressure above that obtained with pure mixtures. 37 − 31 = 6 lbs. per sq. in.
,,	7·1	87·9	5	100	27	
,,	7·1	67·9	25	100	30	
,,	7·1	62·9	30	100	34	
,,	7·1	57·9	35	100	37	
$\frac{1}{13}$	7·7	92·3	none	100	36	43 − 36 = 7 lbs.
,,	7·7	77·3	15	100	42	
,,	7·7	67·3	25	100	43	

The maximum increase found with products of combustion mixtures was in the case of the weakest mixture, and the increase was 19 lbs.

The increase diminished as the proportion of coal gas added ncreased, so that at $\frac{1}{13}$ coal gas the increase was only 7 lbs.

This change is clearly shown below :

Coal gas present	$\frac{1}{16}$	$\frac{1}{15}$	$\frac{1}{14}$	$\frac{1}{13}$	$\frac{1}{11}$
Increase of pressure above corresponding gas and air mixture .	19 lbs.	12 lbs.	6 lbs.	7 lbs.	2 lbs.

The two mixtures containing $\frac{1}{9}$ and $\frac{1}{7}$ of gas showed reduction.

Coal gas present . . .	$\frac{1}{9}$	$\frac{1}{7}$
Deficit of pressure compared with gas and air mixture .	27 lbs.	3 lbs.

The cause of these effects is obvious enough. When a weak mixture, such as that shown in Clerk's experiments with London gas containing $\frac{1}{13}$ of coal gas and $\frac{12}{13}$ air, is ignited, the pressure only rises some 5 lbs. per sq. in., and although there is ample gas present if burned to produce a much higher pressure, yet the gas does not complete its burning—on the contrary, the flame flickers out. If the products of combustion produced from such a mixture be analysed, some 30 per cent. or more of the gas originally put in is found to be still present as inflammable gas. This is found to be the case where the gas was carefully mixed with the air in a separate gasholder, as was done in the Clerk experiments, and where gas is imperfectly mixed a still larger portion may escape combustion. If, then, to products of combustion containing a portion of inflammable gas be added a fresh portion of coal gas and a further supply of air, the mixture contains more gas than that calculated from the added part. This renders it more inflammable, and of course produces a higher pressure.

Consider, first, Grover's experiment with $\frac{1}{16}$ coal gas, or 6·2 per cent., and 93·8 per cent. of air; the explosion pressure attained was only 16 lbs. per sq. in. above atmosphere. When the mixture consisted of the same percentage (6·2 per cent.) of added coal gas, 63·8 per cent. of air, and 30 per cent. of products of combustion, then a pressure of 35 lbs. was obtained upon explosion. If the 30 per cent. of products contained enough gas, unburned, to raise the total percentage of combustible present from 6·2 per cent. to 7 per cent., then, experimenting in Mr. Grover's manner, this pressure would be obtained. The pressure of 35 lbs. would be easily obtained, even allowing for some defects in mixing, as is shown by Clerk's experiments.

The portion of the original gas present in the 30 per cent. would be $6{\cdot}2 \times 0{\cdot}3 = 1{\cdot}9$ nearly, and 0·8 added to 7·2 would produce the necessary 7 per cent., so that this phenomenon is easily explained. As the mixture gets richer and richer in gas less and less gas remains unburned in the combustion products formed till the mixture containing $\frac{1}{11}$ of gas is arrived at, when practically no further unburned gas remains to enrich the added fresh charge in inflammable material.

The author accordingly concludes that it is incorrect to consider that the products of combustion—that is, the mixture of nitrogen, carbonic acid, water vapour, and oxygen—in any way raise the pressures of gaseous explosion or act in any way different from that of a diluent, except in cases when they contain inflammable material remaining from the previous charge.

The author feels it necessary to discuss Mr. Grover's results at some length, as many fallacious explanations have appeared of the action of the exhaust gases, proceeding on the assumption that the gases are inert and uninflammable.

Grover's Later Experiments

Mr. Grover's later experiments on acetylene and air explosions are of a more useful and important kind. They were made between 1898

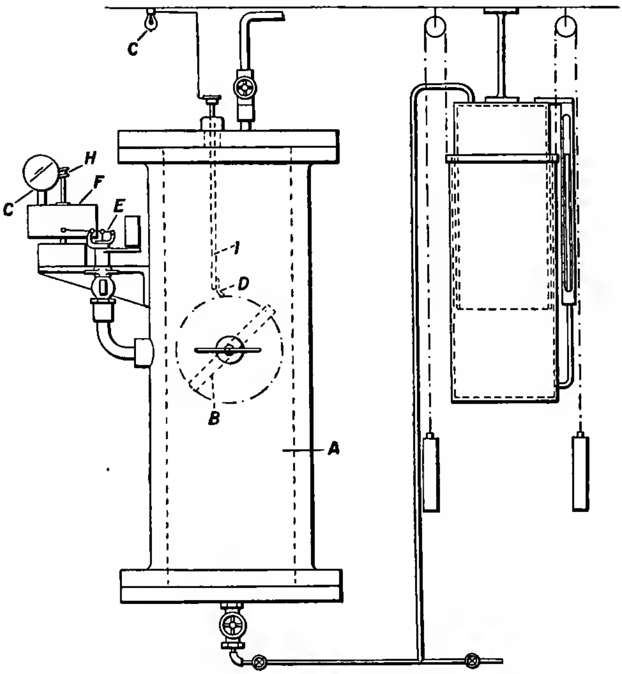


Fig. 47.—Grover's later Explosion Apparatus

and 1901, and the apparatus used corrected many of the errors of his earlier devices. The acetylene was accurately measured from a separate gasholder, and so the introduction of water into the explosion cylinder was avoided. Mr. Grover's later apparatus is shown at fig. 47. It consisted of a vertical explosion vessel A, with covers bolted top and bottom. A rotating wing B was provided, operated by an outside handle on a properly packed spindle. This rotating wing served the double purpose of stirring the mixture first and then igniting by low-tension current. To cause ignition an internal contact breaker I

was coupled in series with a glow-lamp C to the electric light leads of the building. The glow of the lamp clearly indicated that a contact had been made within the explosion vessel, and, by quickly rotating the wing B, which had made contact at D, the circuit was interrupted

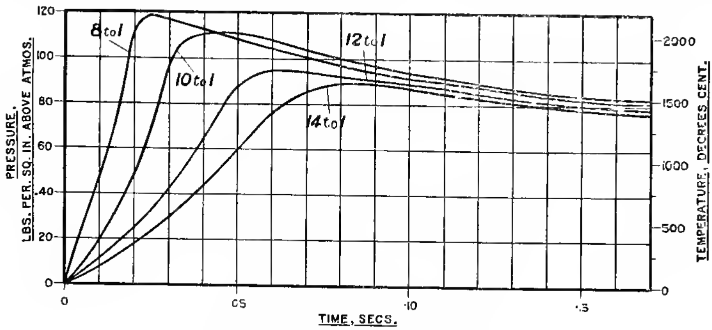


FIG. 48.—Diagrams of Explosion

Various mixtures of acetylene and air ignited at atmospheric pressure. (*Grover*)

and a powerful low-tension spark produced within. A Crosby indicator E was used for determining the pressure, and its pencil was held by a light spring to the revolving drum F driven by clockwork.

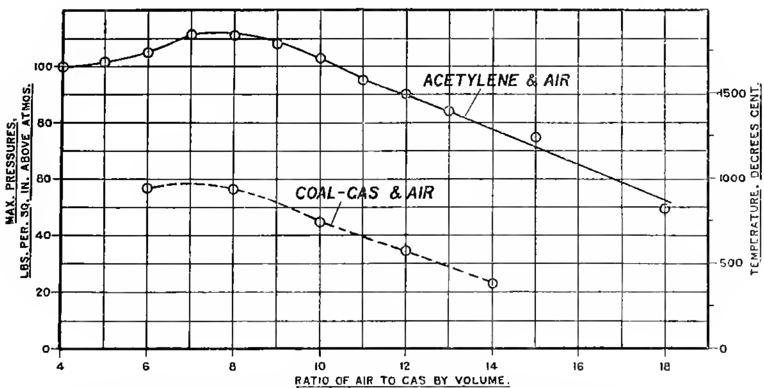


FIG. 49.—Explosion in a Closed Vessel

Maximum pressures recorded when exploding mixtures of acetylene and air, also coal gas and air. Initial pressure, 1 atmosphere; initial temperature, 0° C. (*Grover*)

Mr. Grover timed the revolutions of this drum in an ingenious and effective manner. A watch G was mounted vertically on a small spur-wheel geared to the worm H rotated with the drum, and the arrangement rotated the watch contra-clockwise, so that at a certain speed

the centre-second hand of the watch became stationary. A small mirror was fixed at the centre of the hand, and by observing a distant object as reflected from its surface it was easy to see when the hand was quite stationary. By this device the rotation of the drum could be adjusted with great accuracy to a constant speed for the few seconds of the experiment. Three sets of experiments were made; first, at atmospheric pressure; second, at two atmospheres pressure absolute; and third, at three atmospheres pressure absolute.

Diagrams of acetylene explosions at atmospheric pressure are shown at fig. 48.

The proportion of air to acetylene is marked on each. Temperature scales have been added to Mr. Grover's diagrams.

Fig. 49 shows observations of maximum pressures plotted against the ratio of air to gas and coal gas and air mixtures shown as well.

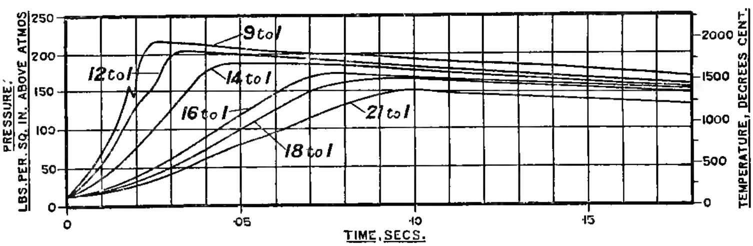


FIG. 50.—Explosion Diagrams of various mixtures of acetylene and air ignited at 2 atmospheres initial pressure; initial temperature, 0° C. (*Grover*)

Mr. Grover gives the following table of pressures produced by explosion, to which the present author has added the times of explosion of mixtures of corresponding strengths taken from the diagram, fig. 48.

MIXTURES OF ACETYLENE AND AIR EXPLODED AT ATMOSPHERIC PRESSURE. (*Grover*)

Initial Temperature, 0° C.

Proportion of air to gas . .	air/gas	18/1	15/1	14/1	13/1	12/1	11/1	10/1	9/1	8/1	7/1	6/1	5/1	4/1
Maximum pressure in lbs. per sq. in. above atmosphere . .		54	74	83	83	89	95	103	108	111	112	106	102	101
Time of explosion in seconds . .				0·085		0·061		0·047		0·024				

Mr. Grover states that no weaker mixture than 18 air to 1 gas could be fired at atmospheric pressure, but that stronger mixtures than 4 to 1 could be fired.

At fig. 49 the coal gas explosions at atmospheric pressure are also plotted and undoubtedly a greatly increased pressure is obtained per unit volume of acetylene consumed. Mr. Grover, however, does not appear to have overcome the difficulty of mixture, as his coal gas figures are undoubtedly too low. Diagrams of acetylene explosions

at two atmospheres initial pressure are shown at fig. 50, arranged, as already described, with reference to the atmospheric pressure diagrams.

Fig. 51 shows the observations of maximum pressure plotted against the ratio of air to gas; coal gas and air mixtures are also shown, and Mr. Grover's table of pressures is given below:

MIXTURES OF ACETYLENE AND AIR EXPLODED AT TWO ATMOSPHERES PRESSURE. (*Grover*)

Initial Temperature, 0° C.

Proportion of air to gas . . . air/gas	21/1	20/1	19/1	18/1	17/1	16/1	15/1	14/1	13/1	12/1	11/1	10/1	9/1	8/1
Maximum pressure in lbs. per sq. in. above atmosphere . .	121	127	115	138	129	143	171	159	170	168	177	166	196	179
Time of explosion in seconds	0·10			0·10		0·08				0·035	0·06		0·025	

At this stage of his experiments Mr. Grover found that certain discrepancies appeared, and, on investigation, it was proved by him

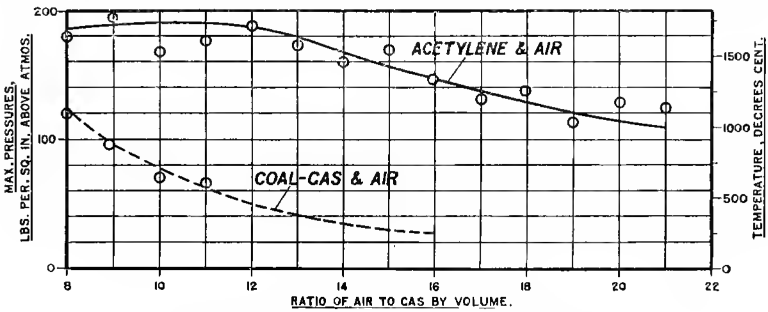


FIG. 51.—Explosion in a Closed Vessel

Maximum pressures recorded when exploding mixtures of acetylene and air, also coal gas and air. Initial pressure, 2 atmospheres; initial temperature, 0° C. (*Grover*)

that some air had got into the gasholder from which he measured his acetylene; so that, instead of measuring pure acetylene from his gasholder, he was in reality sending in a mixture of acetylene and air containing from 6 per cent. to 20 per cent. of air. He considers that 5 per cent. of air is probably present in all the mixtures exploded at an initial pressure of one atmosphere, and probably more at the higher pressures, so that the mixture plotted at fig. 49 as 15 air to 1 gas should in reality be 15·7 air to 1 of gas.

At initial pressure of two atmospheres the weakest mixture which could be fired was 21 gas 1 air as measured by the gasholder. Assuming 10 per cent. of air in the gasholder, Mr. Grover gives the true mixture as 23·1 to 1.

Four mixtures of coal gas in the proportion of 8 to 1 and 11 to 1 were fired at the same initial pressures, and the maximum pressure of

the acetylene explosions was found to be from 1·5 to 2·7 as great as with corresponding mixtures of coal gas.

Diagrams of acetylene explosions at three atmospheres initial pressure are shown at fig. 52, arranged as already described with reference to the two preceding sets of experiments.

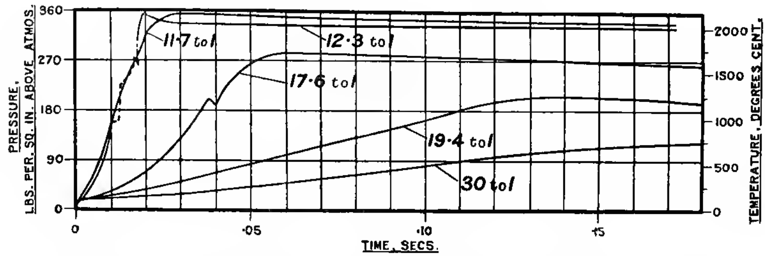


FIG. 52.—Explosion Diagrams of various mixtures of acetylene and air ignited at 3 atmospheres initial pressure ; initial temperature, 0° C. (*Grover*)

Fig. 53 shows observations of maximum pressure plotted against the ratio of air to gas, also coal gas and air mixtures.

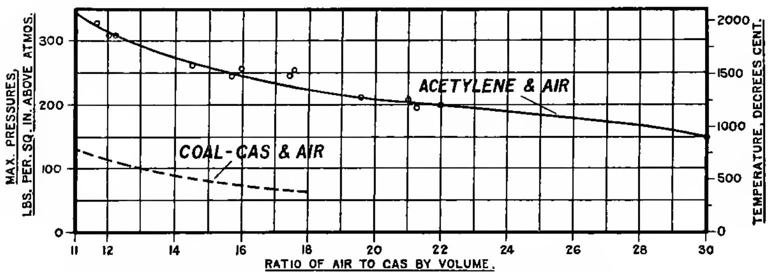


FIG. 53.—Explosion in a Closed Vessel

Maximum pressures recorded when exploding mixtures of acetylene and air, also coal gas and air. Initial pressure, 3 atmospheres ; initial temperature, 0° C. (*Grover*)

Mr. Grover's table of pressures is given below :

MIXTURES OF ACETYLENE AND AIR EXPLODED AT TWO ATMOSPHERES PRESSURE. (*Grover.*)

Initial Temperature, 0° C.

Proportion of air to gas . . .	air	30	22	21	19·6	17·5	16·9	16·1	14·7	12·3	12·1	11·7
	gas	1	1	1	1	1	1	1	1	1	1	1
Maximum pressure in lbs per sq. in. above atmosphere . . .		146	197	207	211	246	236	259	261	308	307	325
Time of explosion		0·20			0·15	0·06				0·02		0·03

In these experiments the strongest mixture fired was 11·7 air to 1 gas, and the weakest 30 to 1. Stronger mixtures were not used,

because the limit of safety of the explosion vessel was being too closely approached.

Mr. Grover points out the great rapidity of explosion of acetylene mixtures as compared with coal gas, and notes that the inertia of the parts of the indicator would cause a material lag, so that the more rapid explosions are more liable to error in the time of explosion. All the experiments at initial pressures higher than atmosphere were conducted as follows: The gas required for combustion was first measured at atmospheric pressure and then driven over into the cylinder; all cocks were then closed, and compressed air was passed into the cylinder until the pressure gauge showed 15 or 30 lbs. as required. The mixture was then stirred and allowed to stand for ten minutes before ignition was attempted. Slight leakage was experienced, but it was allowed for.

As the difficulty of getting pure acetylene gas was found to be great, Mr. Grover determined the proportion of acetylene to air by analysis of the products of combustion of the three-atmosphere explosions, and he believes these experiments to be more accurate than those at two atmospheres.

Grover gives the following table of analyses of the products from the combustion of acetylene and air exploded at three atmospheres:

ANALYSES OF PRODUCTS OF COMBUSTION OF AIR AND ACETYLENE MIXTURES. (*Grover*)

Mixtures $\frac{\text{Air}}{\text{Acetylene}}$	Constituents by volume per cent.					Totals
	CO_2	CO	O	N	Steam. H_2O	
$\frac{11\cdot7}{1}$	13·0	3·2	0·0	79·0	6·5	101·7
$\frac{12\cdot3}{1}$	15·1	0·0	0·0	78·0	8·2	101·3
$\frac{14\cdot5}{1}$	12·4	0·0	2·5	78·9	6·2	100·1
$\frac{16}{1}$	11·8	0·0	4·2	79·0	5·9	100·9
$\frac{17\cdot5}{1}$	10·0	0·0	5·1	79·6	5·0	99·7
$\frac{21}{1}$	8·6	0·0	8·1	78·9	4·3	99·9
$\frac{22}{1}$	9·0	0·0	8·6	79·0	4·5	101·1
$\frac{30}{1}$	6·8	0·0	12·0	78·5	3·5	100·7

The column headed 'Mixtures' has been calculated by Mr. Grover

from the analysis, and the steam has been calculated also from the CO_2, and in one case CO.

Mr. Grover gives the following values, which are useful in making calculations as to acetylene explosions :

Calorific value (C_2H_2) acetylene. *Lower* value 1504 B.Th.U.
,, ,, (C_2H_2) ,, Higher value 1558 ,,

for 1 cub. ft. at 0° C. and 14·7 lbs. per sq. in. absolute.

1 cub. ft. of acetylene (C_2H_2) at 0° C. and 14·7 lbs. weighs 0·0725 lbs.
1 cub. ft. of acetylene (C_2H_2) requires 12·5 cub. ft. of air for complete combustion.

Following *Clerk's* method of determining best mixture as to maximum pressures produced, Mr. Grover gives the numbers below :

BEST MIXTURES FOR MAXIMUM PRESSURE—ACETYLENE AND AIR—AT 1, 2, AND 3 ATMOSPHERES INITIAL. (*Grover*)

		Air	Gas
Initial pressure 1 atmosphere . .	Best mixture	13	1
,, ,, 2 ,, . .	,,	15	1
,, ,, 3 ,, . . .	,,	27	1

Mr. Grover has calculated out the efficiencies of combustion for different acetylene mixtures, using the following specific heat values for mean specific heats from 0° to about 2000° C., as given by Mallard and Le Chatelier.

Steam (H_2O) $= 0·68\ C_v$.
Carbonic acid (CO_2) $= 0·308\ C_v$.
Nitrogen (N) $= 0·215\ C_v$.

He takes oxygen also at 0·308.

Using these numbers he determines the efficiency of combustion as from 47 per cent. to 73 per cent., and he states that these efficiency values are higher for acetylene than has ever been noted for coal gas. With this deduction the author does not agree.

Mr. Grover's experiments are very valuable for discussion, as no other acetylene experiments are available, and they are the more valuable because of Mr. Grover's candid and praiseworthy admissions and details of his difficulties.

PETAVEL'S EXPERIMENTS

Dr. J. E. Petavel, F.R.S., has made many interesting experiments on mixtures of coal gas and air compressed before ignition to pressures of over 1000 lbs. per sq. in., and resulting in explosion pressures of some 9000 lbs. per sq. in. To stand such pressures he has devised an apparatus of a very ingenious kind. It consists of a strong

spherical steel bomb of 4 ins. internal diameter, which has therefore a capacity of 551·9 cub. cm. = 0·0195 cub. ft. The indicator used is of a novel construction, and is thus described by Petavel in a paper published in the 'Philosophical Magazine' in May 1902.

Referring to fig. 54 he states:

'A cylindrical groove is cut half through the walls of the inclosure. The upper part P of the cylinder thus obtained represents the piston of our indicator, and the lower part S the spring. Under the pressure of the explosion the piston P will be forced outwards, a certain small amount corresponding to the elastic compression of the material of which the spring is made. This motion is transmitted to the exterior by the rod R.

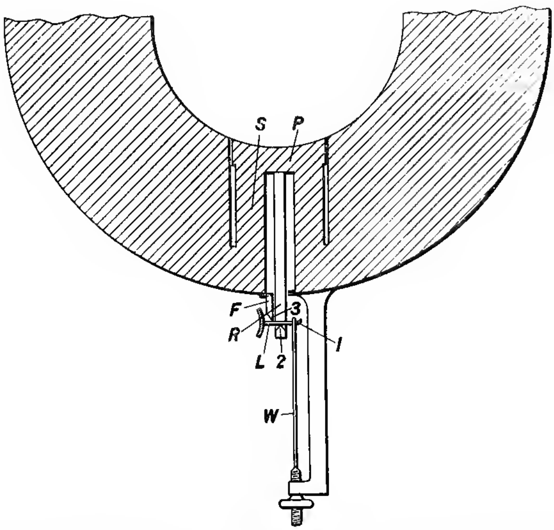


FIG. 54.—Diagrammatic Representation of the Petavel Recording Manometer

'The lever L supporting the mirror rests on the fulcrum F at "3"; it is kept against the knife-edge "2" of R by the tension of the wire W. The wire W is of considerable length, and it is stretched to near its limit of elasticity. The lever L can therefore follow the small advance of the rod R without greatly diminishing the tension of the wire W.

'The mirror focusses a point source of light on to a rapidly revolving cylinder, thus recording on a magnified scale the motion of the piston P. It is not impossible that an indicator of this type would work in practice; but the deflexion of the mirror, and therefore the scale of the records obtained, would be much too small. To increase the deflexions three modifications are necessary: the spring S must be made longer, the ratio of its cross-sectional area to that of the

piston must be decreased, and the knife-edges "2" and "3" brought closer together.

'In fig. 55 the design of the actual instrument is given, the lettering being the same as in the previous figure.

'By means of the thread U the gauge screws into the explosion chamber C being flush with the inside surface; an air-tight joint is formed by the ring D pressing against a flat steel ledge.[1] The end of the gauge from D to E is a good fit in the walls of the explosion chamber, and the joint is thus protected from the direct effect of the explosion.

'The spring S, about 5 ins. in length, is tubular in shape. To

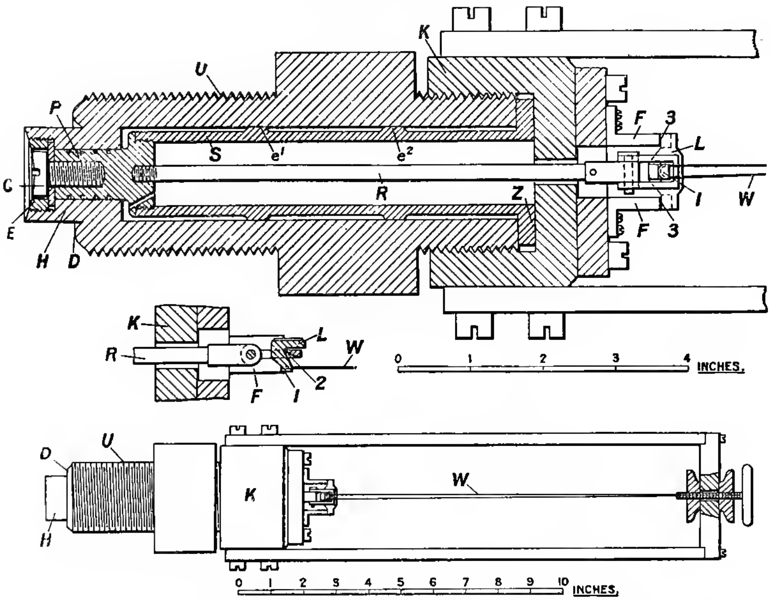


FIG. 55.—Longitudinal Sections and Elevation of the Petavel Recording Manometer

prevent any buckling it is made to closely fit the cylinder in which it is contained at two places, e_1 and e_2. The spring is fixed at the outer end Z, being held in place by the nut R; at the inner end it is free and supports the piston P. The ordinary U-leather is replaced by a leather washer attached to the piston by the screw C and to the fixed part of the gauge by the rim E. The end of the piston projects about an

[1] In the case of apparatus designed for gases under high pressures all joints should be made directly metal to metal, no packing being used. A joint thus made, if properly designed, is and remains absolutely air-tight. It can be made or broken in an instant, and as many times as may be required.

hundredth of an inch above the rim H, and it can therefore move back without straining the leather.

'The mirror (not visible in the figure) is carried by the lever L. This lever is so designed that the knife-edges 1, 2, and 3 are in the same plane, it being at the same time possible to bring the knife-

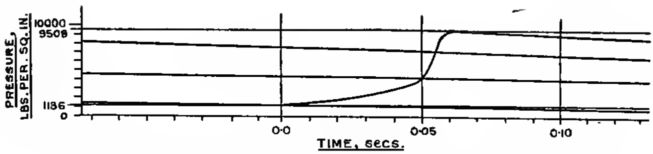


FIG. 56.—Rise of Pressure during Explosion. (*Petavel*)

Spherical enclosure capacity, 551·9 c.c. Temperature of enclosure : before firing, 18° C. ; after firing, 24° C. Initial pressure, 77·3 atmos. (11·36 lbs. per sq. in.). Maximum explosion pressure, 646 atmos. (9,508 lbs. per sq. in.). Ratio $\frac{\text{Air}}{\text{Gas}} = 6·c$. Ratio $\frac{\text{Maximum pressure}}{\text{Initial pressure}} = 8·36$.

edges 2 and 3 within one-hundredth of an inch of each other should so great amplification be found necessary. Up to the present, however, the distance has not been decreased below $\frac{1}{16}$ in., the scale obtained with this distance being found satisfactory.'

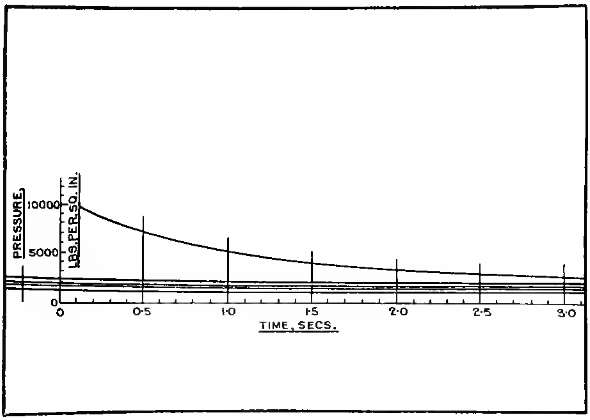


FIG. 57.—Fall of Pressure after Explosion. (*Petavel*)

Spherical enclosure capacity, 551·9 c.c. Temperature of enclosure : before firing, 21° C. ; after firing, 27° C. Initial pressure, 74·38 atmos. (1,094 lbs. per sq. in.). Maximum explosion pressure, 654 atmos. (9,618 lbs. per sq. in.). Ratio $\frac{\text{Air}}{\text{Gas}} = 5·71$. Ratio $\frac{\text{Maximum pressure}}{\text{Initial pressure}} = 8·8$.

The chronograph used is very simple ; it consists of a rotating drum carrying a photographic film. The drum is rapidly rotated by an electric motor, and it is enclosed within a light-tight box having a long, narrow slit ($\frac{1}{32}$ in. wide) running its full length parallel to the

axis of rotation. One of the filaments of an incandescent lamp is focussed on the slit at right angles to it, so as to form a sharp point of light which moves along the slit with increase of pressure.

Figs. 56 and 57 show typical records obtained by Dr. Petavel from mixtures of air and coal gas fired at an initial pressure of about 1100 lbs. per sq. in. Oxygen was in excess, as the residual gases contained about 3 per cent. No part of the instrument except the mirror frame moves more than one or two thousandths of an inch, so that inertia troubles are avoided. Thus, although in fig. 56 the pressure at 0·055 of a second after ignition is rising at the rate of over 1,000,000 lbs. per sq. in. per second, the curve turns sharply at a right angle and shows no sign of vibration at maximum pressure.

Petavel gives the following table with regard to the explosion curve, fig. 56:

Rise of Pressure. (See fig. 56)

Spherical inclosure 4 ins. diameter. Capacity 551·91 c.c. = 0·0195 cub. ft.

Temperature of inclosure before firing, 18° C.
" " after products of combustion had cooled, 24° C.

Initial pressure	77·28 atmospheres = 1,136 lbs. per sq. in.
Ratio: Air to Coal-gas	6·0
Maximum explosive pressure	646·2 atmospheres = 9,508 lbs. per sq. in.
Ratio: maximum explosive pressure to initial pressure	8·4.
Residual pressure	63·8 atmospheres = 937 lbs. per sq. in.

Analysis of Residue.

Carbon dioxide	9·8 per cent.
Oxygen	3·0 "
Nitrogen	87·2 "
	100·0

Time in seconds	Reading in millimetres	Absolute Pressure in lbs. per sq. in.	Time in seconds	Reading in millimetres	Absolute Pressure in lbs. per sq in.
0·000	—	1136	0·058	21·99	9508
0·010	0·77	1237	0·060	21·91	9477
0·020	1·57	1549	0·062	21·91	9477
0·030	3·06	2130	0·064	21·67	9385
0·040	5·03	2898	0·066	21·63	9369
0·042	5·21	2968	0·068	21·60	9357
0·044	5·84	3204	0·070	21·56	9340
0·046	6·38	3424	0·080	21·22	9209
0·048	6·86	3612	0·090	21·07	9149
0·050	7·47	3850	0·100	20·87	9071
0·052	10·80	5147	0·150	19·73	8628
0·054	15·92	7143	0·200	18·61	8192
0·056	19·83	8667			

And this table with regard to the cooling curve, fig. 57:

Fall of Pressure. (See fig. 57)

Spherical inclosure 4 ins. diameter. Capacity 551·91 c.c.
= 0·0195 cub. ft.

Temperature of inclosure before firing, 21° C.
,, ,, after products of combustion had cooled, 27° C.

Initial pressure	74·38 atmospheres = 1,094 lbs. per sq. in.
Ratio: Air to Coal-gas . . .	5·71.
Maximum explosive pressure .	654·2 atmospheres = 9,618 lbs. per sq. in.
Maximum temperature . . .	2483° C.
Ratio: maximum explosive pressure to initial pressure	8·8.
Residual pressure . . .	58·98 atmospheres = 867 lbs. per sq. in.

Analysis of Residue

Carbon dioxide	11·2	per cent.
Oxygen	2·0	,,
Nitrogen	86·8	,,
	100·0	

Time in seconds	Reading in millims.	Absolute Pressure in lbs. per sq. in.	Temperature of Gas in degrees Centigrade *	Time in seconds	Reading in millims.	Absolute Pressure in lbs. per sq in.	Temperature of Gas in degrees Centigrade *
0·08	23·54	9618	2483	2·2	5·68	2979	581
0·1	23·19	9487	2445	2·3	5·54	2927	566
0·2	20·55	8506	2164	2·4	5·32	2845	542
0·3	18·62	7789	1957	2·5	5·13	2774	522
0·4	16·95	7167	1781	2·6	5·00	2726	508
0·5	15·83	6751	1661	2·7	4·83	2663	490
0·6	14·78	6361	1548	2·8	4·68	2607	474
0·7	13·79	5993	1445	2·9	4·51	2544	456
0·8	12·82	5632	1341	3·0	4·35	2484	439
0·9	11·92	5298	1245	3·5	3·82	2287	382
1·0	11·08	4986	1156	4·0	3·33	2094	327
1·1	10·38	4726	1081	4·5	2·96	1968	291
1·2	9·72	4481	1011	5·0	2·72	1878	265
1·3	9·20	4287	955	6·0	2·47	1785	239
1·4	8·68	4094	900	7·0	2·16	1670	205
1·5	8·16	3901	845	8·0	1·95	1592	183
1·6	7·68	3722	793	9·0	1·74	1514	161
1·7	7·30	3581	753	10·0	1·54	1439	139
1·8	6·86	3418	706	11·0	1·43	1399	128
1·9	6·54	3299	672	12·0	1·32	1358	106
2·0	6·23	3183	639	15·0	1·15	1295	98
2·1	5·95	3079	609				

* The temperatures in this column are calculated in the usual manner from the pressure, allowing for the quantity of water-vapour formed.

From the last table he has prepared the cooling curve shown at fig. 58.

Petavel calls attention to the three following points shown by his experiments :

'1. The time required to reach the maximum pressure, namely, 0·058 second, is not far from that which would be required with the same mixture at atmospheric pressure.

'2. The ratio of explosive to initial pressure has been increased. At or near atmospheric pressure the ratio for this mixture would be about 7 ; in the present case it is 8·6. This fact is due to three causes which work simultaneously, namely: (A) The departure of gases from Boyle's law ; (B) the relative decrease of thermal loss during the time occupied by the combustion ; (C) the increase in the absolute temperature at which dissociation would take place.

'3. The rate of cooling has greatly decreased.'

Petavel also states :

'The quantity of heat dissipated per unit of cooling surface increases with the temperature interval and with the pressure of the gas, but not at the same rate as the latter. The heat developed, on the other hand, is simply proportional to the pressure.

'By increasing the pressure from 1 to 70 atmospheres we increase

the heat generated in a given volume 70 times, but we do not increase the rate at which heat is dissipated in anything like the same ratio.'

Royal College of Science Experiments

These experiments were proposed by Professor Perry, and the main part of the apparatus was designed by Messrs. M'Diarmid and Mann, students of the Royal College of Science, South Kensington. Messrs. Leonard Bairstow, A.R.C.Sc., and A. D. Alexander, A.R.C.Sc., made

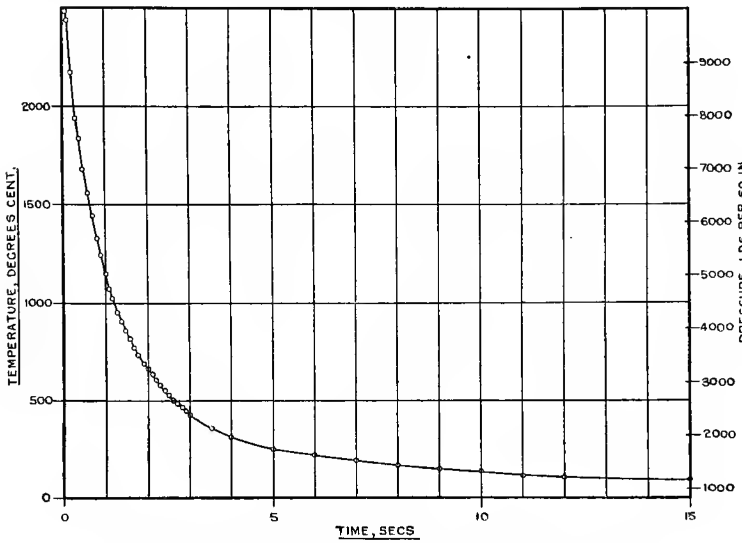


Fig. 58.—Coal Gas and Air. Rate of Cooling. (*Petavel*)

Initial pressure .	1,094 lbs.
Explosion pressure . .	9,618 lbs.
Ratio $\frac{\text{Air}}{\text{Gas}}$	5·7
Ratio $\frac{\text{Explosion pressure}}{\text{Initial pressure}}$.	8·8

the experiments with the sanction and encouragement of Professor Perry. The work occupied two years continuously, and a paper was written by Messrs. Bairstow and Alexander, which was read at the Royal Society in 1905. Only part of the paper was published by the Royal Society, the part which was considered most interesting from the purely scientific standpoint, and the present author is accordingly indebted to Messrs. Bairstow and Alexander and the Royal Society for a copy of the complete paper with its numerous tables illustrating many experimental points.

Much of the work is of moment to the engineer, and the author has extracted the parts of the greatest importance.

The apparatus used is illustrated diagrammatically at fig. 59. A is an indicator of the Simplex type, mounted upon an explosion vessel and having a rotating drum operated mechanically and connected to a striker B, which served to enable the apparatus to be timed to the beats of a metronome, so that in all experiments 42·5 ins. of diagram represented one second in time; C is the driving cord which served to actuate the striker and drum; D is a Bourdon gauge, which indicated the initial pressure before explosion; E a temperature plug containing mercury to get the initial temperature of the apparatus before firing; F is a firing tube, to be described more fully later; G is an

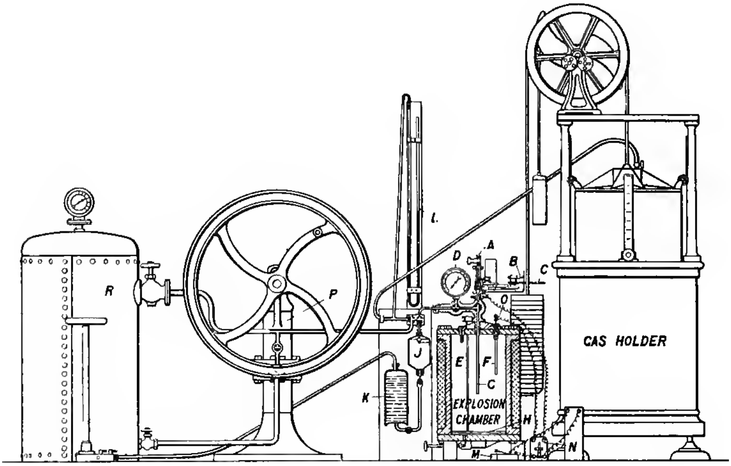


FIG. 59.—Diagrammatic View of Royal College of Science Explosion Apparatus

indicator tube, which also serves as a charging tube; H is a perforated stirring plate, operated from the exterior by a rod and handle shown, used to mix the gases thoroughly before firing; J, K is the gas-measuring device, to enable the gas to be measured at atmospheric pressure and discharged into the explosion vessel under pressure; L is a mercury gauge; M is a sparking key; N the induction coil, and O a sparking plug.

The explosion chamber is 18 ins. long by 10 ins. internal diameter, and its capacity is 1419 cub. ins. It is of cast iron, and strong enough to stand 1000 lbs. per sq. in.

A pump P supplies air to an air reservoir R, and this air is admitted at the desired pressures to the explosion chamber by way of the indicator cock and suitable pipe connections.

It was found at an early stage of the experiments that mixing of the gas and air required special attention; both maximum pressure and time of explosion varied greatly for identical proportions of gas, in accordance with the perfection or imperfection of mixing. Thus, with a mixture of 0·225 gas to 1 volume of air, the maximum pressure varied 30 per cent. when the gas was admitted to the air at the top of the cylinder, depending on the rate at which it was caused to flow into the air. In one experiment the gases were left for seventeen hours, in the hope that gaseous diffusion would produce a uniform mixture, but even then ignition could not be produced by a spark passed at the top of the cylinder; this proved that the mixture then was too rich for ignition and had not diffused throughout the vessel. By adding air rapidly to the gases, however, very consistent results were obtained. It was found necessary to add the mixing plate H (fig. 59), and the method of charging used for the experiments was as follows: The explosion cylinder was put into connection with the suction pump and the mercury gauge, and the pressure reduced below atmosphere to a known amount as measured by the mercury gauge; the suction pump was disconnected and the cylinder put into connection with the gasholder, and gas was admitted to bring the pressure within up to one atmosphere; the mercury gauge was then disconnected, and communication opened with the air reservoir till the pressure rose to 35 lbs. per sq. in. above atmosphere. The stirrer was then operated to thoroughly mix the whole of the gases. The pressure in the vessel was then reduced to the initial pressure required, and the reading of the initial pressure was made on the Bourdon pressure gauge, which was carefully calibrated. The initial temperature of the explosion chamber was then read, and all the instruments were then disconnected. The recording drum was set in motion, the indicator was opened to the cylinder, and the spark passed within the next half-second. After the cooling of the products of combustion to nearly the initial temperature the pressure in the cylinder was read. The cylinder was cleared of the products of combustion by filling up with air twice and blowing off at the top and bottom alternately.

When the experimenters had satisfied themselves that they always obtained a homogeneous mixture in the explosion vessel, they made preliminary experiments to discover if the position at which firing was started within the explosion chamber made any difference as to time of explosion and maximum pressure attained. For this purpose they used the firing tube F; this tube was plugged at the lower end, but it had a pin-hole aperture in the side directed horizontally. When the gases were ignited within the firing tube by the spark, the position of the pin-hole determined the point within the vessel from which ignition started. Taking one mixture in this way and firing with the

pin-hole at different distances from the upper cover of the explosion chamber, the explosion was found to be most rapid fired at the middle point of the chamber.

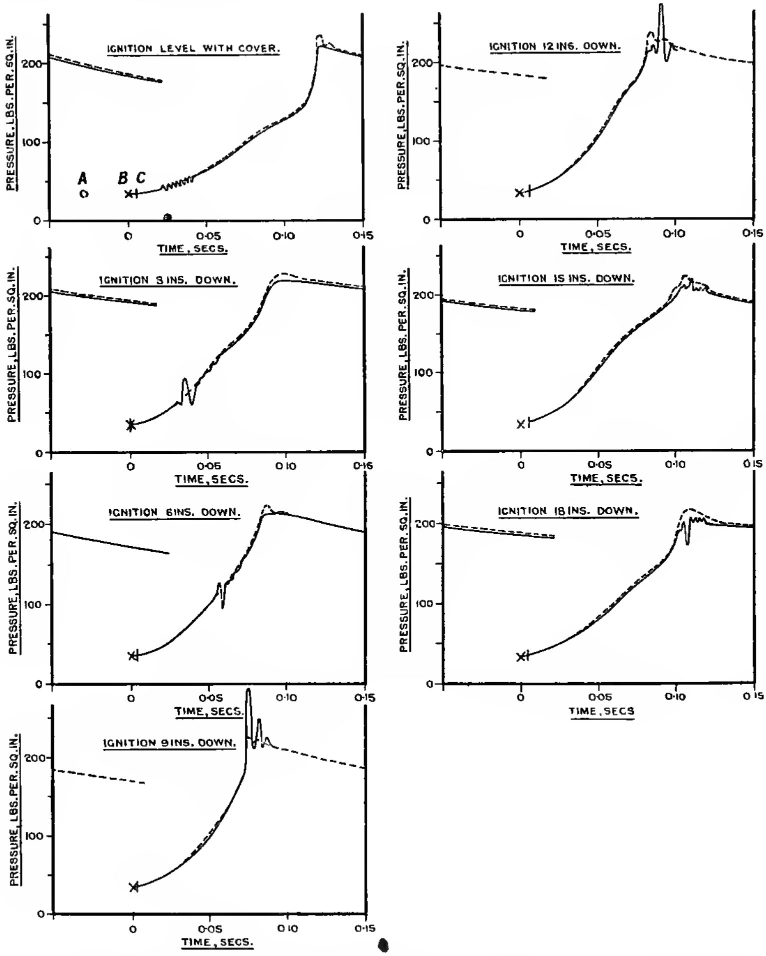


FIG. 60.—Explosion Diagrams from a mixture of 1 air and 0·142 coal gas, with initial pressure 35 lbs. per sq. in. absolute and ignition started at varying distances from the upper cover. (*Bairstow and Alexander*)

The time of explosion was 0·07 second less when the mixture was ignited at the middle of the vessel 9 ins. from the cover, than when ignited 1·2 in. from the cover. Notwithstanding this difference in time the maximum pressure attained in both cases was the same.

The mixture was rich in gas—1 air to 0·125 gas—so that the explosion was rapid. The authors conclude that, as maximum pressures coincide, the cooling loss during explosion must be small.

In the case of a weak mixture, when ignited near the top, the maximum pressure of 137 lbs. per sq. in. absolute was attained in 1·8 second; when ignited 12 ins. from the top, a maximum pressure of 180 lbs. per sq. in. absolute was attained in 0·4 second. The cooling action of the cylinder during the explosion period (1·8 second) prevents an increase of more than 30 per cent. of the pressure (137 lbs.) actually attained.

The diagrams obtained with a rich mixture, 1 air and 0·142 of gas —that is, very nearly 7 of air to 1 of gas—are shown at fig. 60.

The initial pressure was 35 lbs. per sq. in. absolute, the mixture and initial temperature was the same in all cases. The mixture was fired with the pin-hole, level with the inside of the upper cover, 3 ins., 6 ins., 9 ins., 12 ins., 15 ins., and 18 ins. down. The latter figure, of course, means that the pin-hole was at the bottom of the vessel. Two diagrams were simultaneously taken by two indicators; one of the diagrams is shown in full lines, the other in dotted lines.

X B indicates the point where the full-line diagram begins to be recorded and I C the same point for the dotted line.

In the first diagram of the set O A marks the real point in time when the electric spark passes as determined by a spark in series, passing through the indicator paper as well as the sparking points within the cylinder.

The true time of explosion is obviously the time elapsing between the passage of the spark and the attainment of maximum pressure.

This is given in the following table, together with the maximum pressures attained:

Depth of pin-hole in firing tube from top of vessel	Time from spark to maximum pressure	Maximum pressure of explosion
ins.	sec.	lbs. per sq. in. abs.
0	0·160	220
3	0·141	220
6	0·117	220
9	0·105	223
12	0·123	223
15	0·140	218
18	0·145	218

From these experiments it is seen that explosion is most rapid when the mixture is fired at the middle of the vessel. The maximum pressure is then attained in 0·105 second after the moment of passing the spark; fired at the top, this time is 0·160 second, and at the bottom

0·145 second. The maximum pressure, however, varies but little: 220 lbs. absolute with the top firing, 223 lbs. with the middle firing, and 218 lbs. with the bottom firing.

Evidently it is best to fire at the middle position of the vessel in these experiments when the most rapid ignition is desired.

Further experiments with the same composition of mixture proved that with top firing the indicator lagged behind the spark in indicating beginning of pressure rise by approximately 0·03 second—that is, three-hundredths of a second.

By arranging, however, for four sparks in series to pass right through the axis of the cylinder the indicator lag was reduced to 0·017 second, or about half the lag given when one spark was used at the top of the cylinder.

Similar experiments were made with a weak mixture, fired in the same manner at different points in the cylinder. The initial pressure was 50 lbs. per sq. in. absolute.

The results are given in the following table :

Depth of pin-hole in firing tube from top of vessel	Time from spark to maximum pressure	Maximum pressure of explosion
ins.	sec.	lbs. per sq. in. abs.
2	1.9	152
4	1·1	168
6	0·8	178
8	0·8	173
10	0·65	189
12	0·50	197
14	0·55	197
16	0·60	194

Here the shortest time of explosion was found when the igniting point was 12 ins. from the upper cylinder cover and the longest time was at 2 ins. below that cover. The shortest time of explosion was 0·5 second and the longest 1·9 second. The lowest maximum pressure was 152 lbs., and the highest 197. It is to be remembered that all these changes occur with identical mixtures, the only difference being the point of firing in the vessel. With this slow explosion the best point for firing is 3 ins. lower than the centre of the vessel, because convection is sufficiently rapid to be comparable with the rate of combustion and the lower position assists the action of convection.

In these experiments the indicator lagged behind the spark from 0·25 second at 2 ins. from top ignition to 0·11 second with the 12-in. position.

With a similar weak mixture the four-spark method reduced the lag to 0·036 second.

All the experiments now to be described were made using four

sparks in series, passing through the axis of the cylinder and also using mechanical mixing by the plate.

Two sets of experiments were made with varying mixtures—one set at an initial pressure of 55 lbs. per sq. in. absolute, and the other at 34·5 lbs.

Fig. 61 shows the explosion and part of the cooling curves taken with varying mixtures of gas and air at an initial pressure of 55 lbs. per sq. in.

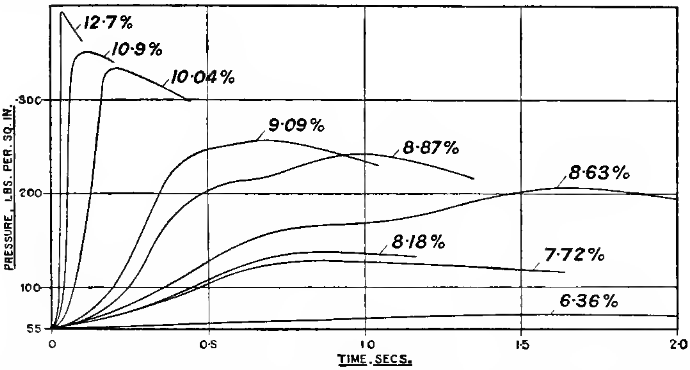


FIG. 61.—Explosion and part Cooling Curves of various mixtures of gas and air at initial pressure of 55 lbs. per sq. in. absolute. (*Bairstow and Alexander*)

The composition of the mixture is marked on each curve, and it will be seen that the mixture varies from a gas content of 6·36 per cent. to 12·7 per cent.

The following table gives the numbers relating to fig. 61 :

EXPLOSION IN A CLOSED VESSEL. (*Bairstow and Alexander*)

Initial pressure . . . . 55 lbs. per sq. in. absolute
Initial temperature . . . . . . atmospheric
Mixture containing from 6·36 to 12·7 per cent. gas

Fraction of total volume occupied by gas	Maximum pressure observed lbs. per sq. in. abs.	Time to reach maximum pressure in seconds	Maximum temperature of explosion °C.
Per cent.			
6·36	70	1·5 to 1·8	—
7·27	—	—	—
7·72	127	0·8	396
8·18	140	0·8	465
8·63	208	1·7	823
8·88	239	1·0	790
9·09	255	0·7	1070
10·04	331	0·2	1470
10·09	350	0·1	1570
12·07	385	0·04	1760

Fig. 62 shows the explosion and part of the cooling curves taken with varying mixtures of gas and air at an initial pressure of 34·5 lbs. per sq. in. absolute.

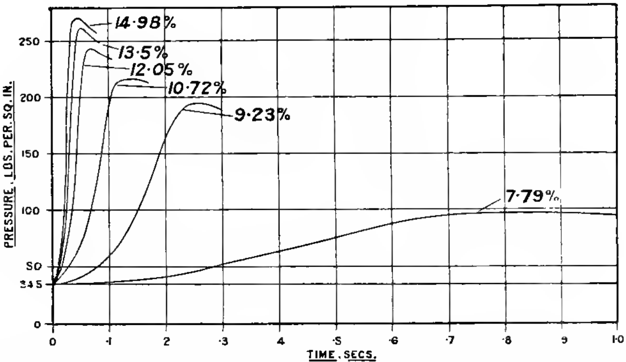


FIG 62.—Explosion and part Cooling Curves of various mixtures of gas and air at initial pressure of 34·5 lbs. per sq. in. absolute. (*Bairstow and Alexander*)

The composition of the mixture is marked on each curve, and it will be seen that the mixture varies from 7·79 per cent. gas to 14·98 per cent.

The following table gives the numbers relating to fig. 62 :

EXPLOSION IN A CLOSED VESSEL. (*Bairstow and Alexander*)

Initial pressure . . . . 34·5 lbs. per sq. in.
Initial temperature . . . . 17° C.
Mixture containing from 7·79 per cent. to 14·98 per cent. gas

Fraction of total volume occupied by gas	Maximum pressure observed, lbs. per sq. in. above atmos.	Time to reach maximum pressure in seconds	Maximum temperature of explosion, °C.
Per cent.			
7·79	97	0·8	540
9·23	193	0·24	1350
10·72	216	0·13	1540
12·05	245	0·07	1790
13·5	263	0·05	1930
14·98	272	0·04	2010

Two sets of experiments were made with varying initial pressures and constant mixtures. In both sets the initial pressure was varied from about 7 lbs. to 45 lbs. per sq. in. absolute.

One mixture contained approximately 14·4 per cent. of gas, and the other 9·5 per cent.

Fig. 63 shows the explosion and part of the cooling curves of the stronger mixture.

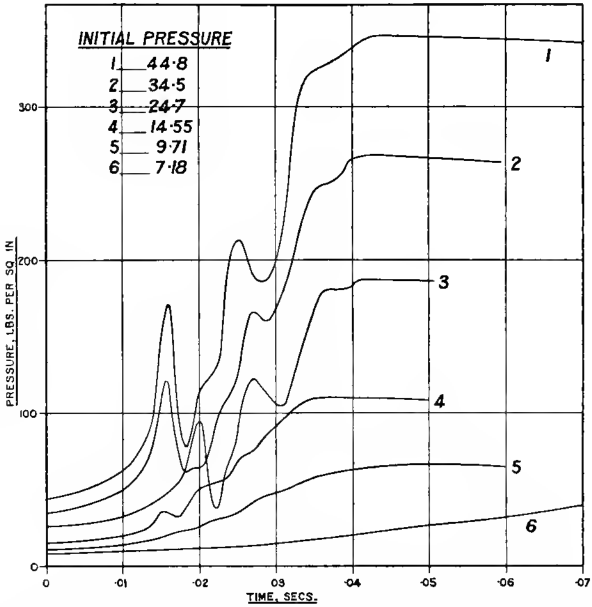


FIG. 63.—Explosion Curves of gas and air mixture containing 14·4 per cent. of coal gas at varying initial pressures between 7 and 45 lbs. per sq. in. absolute. (*Bairstow and Alexander*)

The following table gives the numbers relating to fig. 63 :

EXPLOSION IN A CLOSED VESSEL. (*Bairstow and Alexander*)

Initial pressure varied from 7·18 lbs. to 44·8 lbs. per sq. in. absolute
Initial temperature approximately . . . . 22·5 C.
Mixture containing 14·4 per cent. gas and 85·6 per cent. air

Mixture.		Initial pressure, lbs. per sq. in. abs.	Initial temp. °C	Max. pressure observed lbs. per sq. in. abs.	Time to reach max. pressure in seconds	Max. temp. of explosion °C.
Air	Gas					
1	0·169	44·8	23·5	348	0·042	2030
1	0·172	34·5	22	270	0·041	2040
1	0·170	24·7	21	189	0·041	1980
1	0·168	14·55	21	112	0·036	1990
1	0·166	9·71	24·5	68	0·05	1810
1	0·166	7·18	24	47	0·10	1670

Fig. 64 shows the explosion and part of the cooling curves of the weaker mixture :

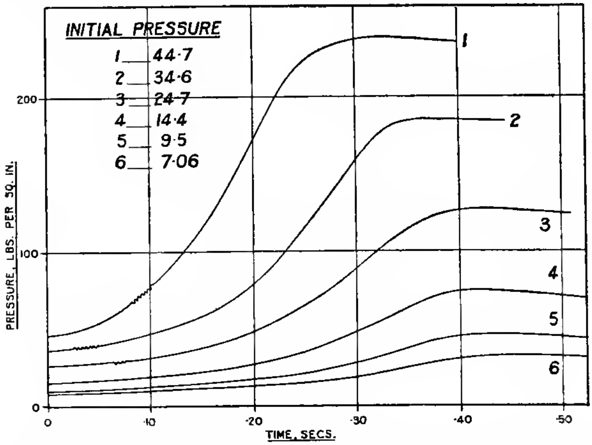


FIG. 64.—Explosion Curves of gas and air mixture containing 9·5 per cent. of coal gas at varying initial pressures between 7 and 45 lbs. per sq. in. absolute. (*Bairstow and Alexander*)

The following table gives the numbers relating to fig. 64 :

EXPLOSION IN A CLOSED VESSEL. (*Bairstow and Alexander*)

Initial pressure varied from 7·06 lbs. to 44·7 lbs. per sq. in. absolute.
Initial temperature . . atmospheric.
Mixture containing 9·5 per cent. gas and 90·5 per cent. air.

Mixture		Initial pressure, lbs. per sq in. abs.	Initial temp. °C.	Max. pressure observed, lbs. per sq. in. abs.	Time to reach max. pressure in seconds	Max. temp. of explosion °C.
Air	Gas					
1	0·103	44·7	16·5	238	0·33	1270
1	0·105	34·6	18·6	185	0·35	1280
1	0·104	24·7	20·0	126	0·41	1220
1	0·107	14·4	21·0	74	0·44	1235
1	0·104	9·5	21·5	46	0·50	1150
1	0·107	7·06	22·0	33	0·50	1110

The cooling curves for the last two sets of experiments are shown at fig. 65.

In this figure an arbitrary zero line of time is shown, so that all the curves begin at the same temperature at the small circles shown. The second circles on the cooling lines also show equal temperatures, and from these it is obvious that the higher the initial pressure of the

mixture the more slowly does the temperature fall. These curves are important and will be discussed later.

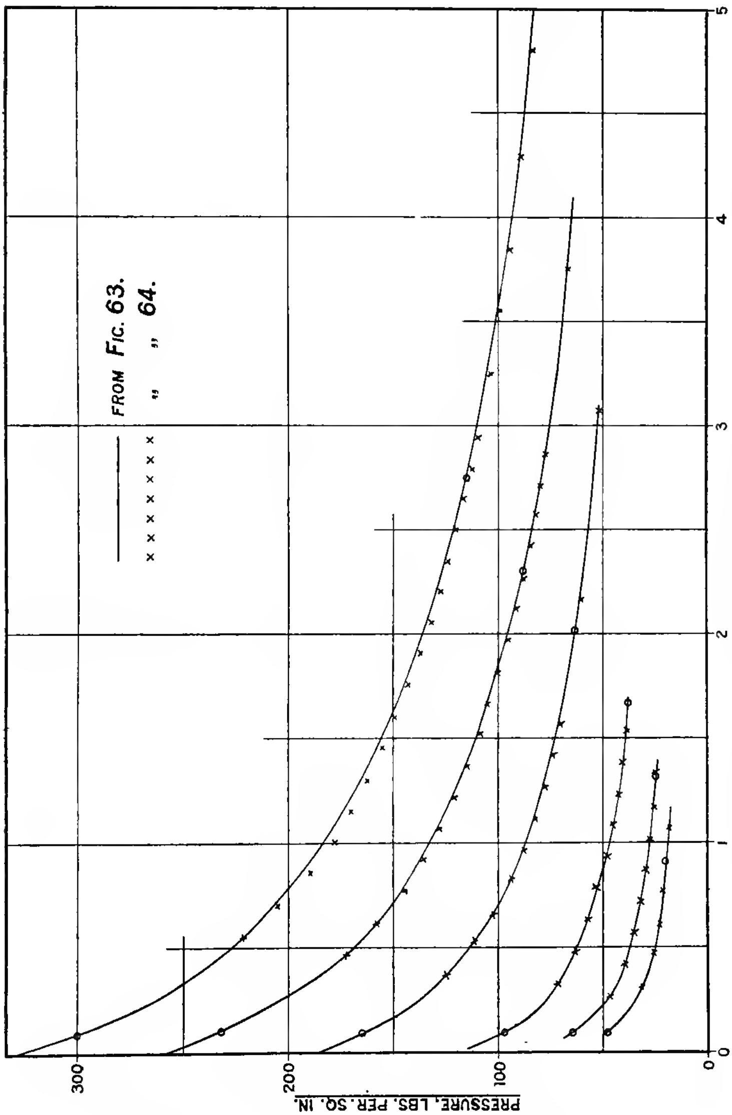


FIG. 65.—Cooling Curves for Explosion Experiments shown at figs. 63 and 64. (*Bairstow and Alexander*)

The composition of the coal gas used in these experiments is given by Messrs. Bairstow and Alexander as follows, from their analysis :

ANALYSIS OF COAL GAS USED IN BAIRSTOW AND ALEXANDER'S EXPERIMENTS

	Per cent. per volume
Marsh gas, CH_4	27·8
Hydrogen, H	42·8
Heavy hydrocarbons, $C_{3 \cdot 23}H_{11 \cdot 9}$	5·4
Carbonic oxide, CO	11·5
Oxygen, O	0·1
Nitrogen, N	12·0
	99·6

An analysis by the Gas Company of earlier date gives the heavy hydrocarbons at $C_{2 \cdot 85}H_{9 \cdot 53}$. The higher heating value of this gas as measured by a Dowson calorimeter was 682 B.Th.U. or 379 Centigrade heat units; but determinations at other times showed 648 B.Th.U. or 360 Centigrade heat units higher value for 1 cub. ft. of the dry coal gas measured at 0° C. and 30 ins. mercury.

Bairstow and Alexander have calculated the proportion of heat accounted for at maximum temperature of explosion for mixtures containing 0·184 gas to 1 of air (1 volume gas, 5·3 volumes air), and 0·102 gas to 1 of air (1 volume gas 9·8 volumes air), using the following values:

Calorific value (higher), 1 cub. ft. of coal gas at 0° C. and 30 ins. mercury . . . . . = 379 Centigrade heat units.
= 682 B.Th.U.

Specific heat at constant volume of 1 cub. ft. of oxygen or nitrogen at 0° C. and 30 ins. mercury . = 0·0138 Centigrade heat units.
„ „ Carbon dioxide = 0·0211 „ „ „
„ „ Steam = 0·0197 „ „ „

For the mixture 1 volume gas, 5·3 volumes air, the composition of the products of combustion was:

CO_2 = 0·216 cub. ft.
N = 1·677 „
O = 0·013 „
Water of saturation, H_2O = 0·018 „
Volume of steam formed, H_2O = 0·498 „

In this experiment 0·382 cub. ft. of coal gas was burned from an initial pressure of 44·1 lbs. per sq. in. absolute, giving a maximum pressure when corrected for cooling during explosion of 403 lbs. per sq. in. absolute.

If the maximum pressure of complete heat evolution be calculated from the above figures, it should have been 591 lbs. per sq. in. absolute.

That is, the heating value of the gas measured by the rise of pressure corrected for cooling is 65·7 per cent. of the calculated value, and the corresponding temperature attained is 2410° C. instead of 3660° C., as calculated.

With the mixture containing 1 gas to 9·8 of air the rise of pressure is 80 per cent. of the calculated value.

Bairstow and Alexander state that these values are higher than those previously noted, because

a. The specific heat values adopted by them are somewhat greater.

b. The actual maximum pressure obtained is greater.

c. A cooling correction is applied.

Bairstow and Alexander also state :

'The determination of the actual maximum temperature and the amount of heat developed at that point involves considerable difficulties.'

Mr. D. Clerk discusses the effect of change of volume on the temperature, and says : 'It is evident that the limits of possible error in calculating temperature from pressure of explosion does not exceed, in the worst case, with coal gas and air 3·4 per cent., and in weaker mixtures half that number.' The 3·4 per cent. referred to is the difference between the volumes before and after explosion on the basis of no condensation of water vapour. This statement is fundamentally incorrect; if the products of combustion were heated to such a temperature that carbon dioxide was completely dissociated into carbon monoxide and oxygen and the steam into hydrogen and oxygen, the volume would correspond to one 12 per cent. greater than the volume of coal gas and air before explosion, and the contraction which is really concerned in the calculation of temperature is therefore more than 15 per cent., and the uncertainty is proportionately increased.'

The present author does not agree with Messrs. Bairstow and Alexander in their criticism of his statement of the case made in 1886. The question of the change of volume caused by dissociation of carbon dioxide and of steam is dealt with in the Report of the Committee of the British Association on Gaseous Explosions, which will be found in Appendix IV.

Hopkinson's Experiments with a Large Explosion Vessel

Important experiments have been made by Prof. Hopkinson, of Cambridge University, and published in a paper read before the Royal Society in February 1906. They are important in three respects : the explosion vessel was very large, its capacity was 6·2 cub. ft., and it

was of dumpy cylindrical form, as shown in section at fig. 66. 59·5 cm. diameter (23·4 ins. diameter), 73 cm. long (28·75 ins. length); pressure changes were measured by an optical indicator; temperatures at various parts of the vessel were measured by fine platinum-wire resistance thermometers placed at different positions in the vessel.

Experiments were made with two mixtures—1 of gas to 9 of air; and 1 of gas to 12 of air—that is, mixtures containing respectively 10 per cent. and 7·7 per cent. of gas. In all experiments it was endeavoured to saturate the mixture with water vapour.

The experiments were all made with the initial pressure atmosphere: with a mixture of Cambridge coal gas having an average 'higher' calorific value of 378 Centigrade heat units or 680 British thermal units per cubic foot at 0° C. and 760 mm. mercury.

The analysis of Cambridge gas, together with oxygen required for combustion, steam, and carbonic acid produced is as follows:

Analysis of Cambridge Coal Gas. (*Hopkinson Experiment*)

—	Per cent. by volume	Oxygen required for combustion	Steam produced	Carbonic acid produced
Hydrogen, H . .	47·2	23·6 vols.	47·2 vols.	—
Marsh gas, CH_4 . .	35·2	70·4 vols.	70·4 vols.	35·2 vols.
Heavy hydrocarbons .	4·8	22·6 vols.	16·0 vols.	14·4 vols.
Carbonic oxide, CO .	7·15	3·6 vols.	—	7·15 vols.
Nitrogen, N . .	5·4	—	—	—
Other gases . .	0·25	—	—	—
	100·00 vols.	120·2 vols.	133·6 vols.	56·75 vols.

By experiment, 100 volumes of gas require 576 volumes air for complete combustion.

By experiment, 100 volumes of gas burned in 900 volumes air give 133 volumes steam, 57 volumes CO_2, and 780 inert gases.

By experiment, higher calorific value of 1 cub. ft. at 0° C. and 760 mm. mercury = 680 B.Th.U. or 378 Centigrade heat units.

Referring to the section of the explosion vessel, fig. 66, A is the sparking-point, nearly at the centre of the vessel; B, C, and D are three platinum thermometers. B is close to the spark, C is 30 cm. from the spark, and D is 1 cm. from the walls of the vessel.

Each thermometer consists of a coil of about 5 cm. of pure platinum wire of 0·001 in. diameter.

Glass tubes carry stout copper leads to the platinum wires, and the copper-platinum junctions are made within the glass tubes so as to be protected from the flame. Each thermometer coil is placed in series with a storage cell and with a d'Arsonval galvanometer having a stiff

phosphor-bronze suspension giving a period of from $\frac{1}{50}$ to $\frac{1}{30}$ of a second. The mirror of the galvanometer throws the image of a fine hole, illuminated by an arc lamp, on a revolving drum, carrying a photographic film.

Knowing the resistance of the platinum wire when cold, its rise in temperature can be calculated. Usually two thermometers were in use at once and a record was obtained from both, on the same drum and in the same explosion, of the changes of temperature at two different points of the vessel. A record of pressure change was taken on the same drum. The indicator was of simple construction, consisting of a steel piston, which was forced by the pressure against a piece of straight spring held at the ends. The displacement of the

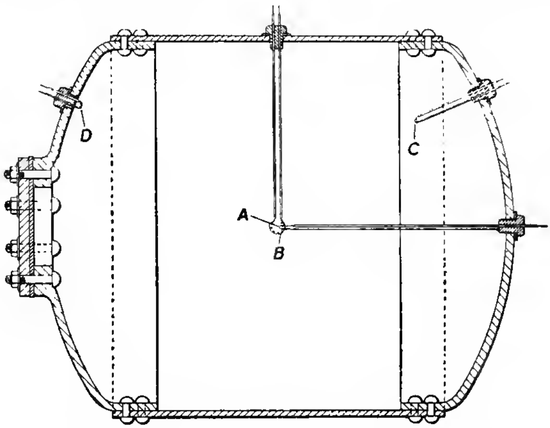


FIG. 66.—Hopkinson Explosion Vessel

spring tilted a mirror about a fulcrum, and the mirror cast an image of the above-mentioned fine hole on the moving film.

The indicator had a natural period of about $\frac{1}{300}$ of a second, and so was able to follow the changes of pressure with great rapidity.

The experiments were made as follows: Steam was blown into the vessel, so that the air should be as nearly saturated with moisture as possible. The vessel was then exhausted, say, to $\frac{9}{10}$ of an atmosphere, and gas was admitted to bring the pressure up to atmosphere; this gave a mixture of 1 gas to 9 air. The explosion vessel was allowed to stand from four to six hours to permit mixing by gaseous diffusion. The completeness of combustion was tested in a few cases by measuring the fall of pressure due to the condensation of steam formed by the explosion. The combustion was found to be satisfactorily complete. Experiments were made to determine the lag of the platinum thermometer behind the temperature of the gases.

From this short account it will be seen that Prof. Hopkinson's experiments were very carefully made, and all precautions taken to obtain accuracy. The only uncertain point in the present author's view is in the question of mixing—even a six hours' delay may not have completed diffusion. With the large vessel used, however, and the relatively small gas-inlet aperture, it is probable that the velocity of the entering gas was sufficient to obtain good mixing. The diagrams obtained seem to be those proper to a good mixture.

Explosion of Rich Mixture, 1 Gas, 9 Air

With 9 volumes of air to 1 of gas the maximum pressure of explosion varied from 76 to 82 lbs. per sq. in. above atmosphere, and was reached in 0·25 second after firing.

Fig. 67 is a copy of the photographic diagram produced, with its pressure line and two temperature lines.

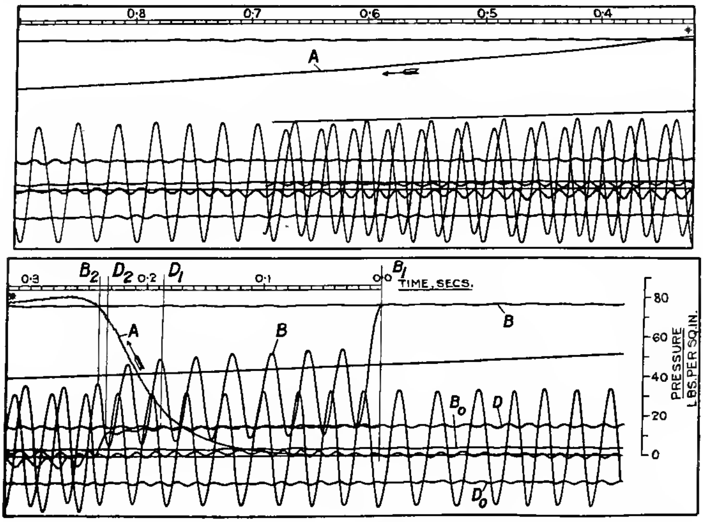


Fig. 67.—Explosion and Cooling Curve of gas and air mixture (1 gas and 9 air) at atmospheric pressure, with temperatures reached by platinum resistance thermometers. (*Hopkinson*)

The film is shown in two parts: A is the pressure curve, of which the explosion portion is shown on the lower part; the zero point of the horizontal line of time indicates the point of passing the spark; the line of cooling is continued on the second part, as indicated by the asterisk.

B is the curve recording the temperature determined by the centre thermometer B (fig. 66), and B_0 is the zero line for this curve, traced on the film immediately after the explosion by disconnecting the

thermometer circuit. The current flowing in the circuit is, therefore, after allowing for inertia effects, proportional to the ordinate of the curve B, reckoned from the zero line B_0, and the resistance of the circuit is inversely proportional to that ordinate.

D is the curve recording the temperature determined by the thermometer D (fig. 66), which is 1 cm. from the side. The galvanometer here has a stiffer suspension and a lower period of oscillation, so that it is less sensitive. D_0 is the zero line for this record.

On this diagram we have, therefore, the means of determining by direct measurement the temperature changes occurring at the centre of the vessel close to the point of starting the explosion, the temperature at 1 cm. from the wall of the vessel at the same time, and the changes of pressure shown by the indicator, giving mean temperature of the whole mass of contained gas.

Comparing first the indications of thermometer B with the explosion line of the pressure indicator at the point B_1, the curve B rapidly departs towards the zero line B_0, showing a reduction in resistance, and consequently a rise in temperature. The galvanometer is thrown into violent oscillation by the rapidity of the temperature change. This indicates the contact of the flame which has been started on its course by an electric spark passed at the point A, about 2 cm. distant. This is taken as the beginning of the explosion, and it will be seen that it has had so far no effect on the pressure in the vessel as indicated by the pressure line A. Smoothing out the oscillations of the curve B, it will be seen that the temperature remains nearly constant, there is a very slow rise, till the point B_2 is reached, when a further abrupt temperature rise occurs, so that the curve falls to the zero line B_0; it continues oscillating about that line for the remainder of the record. The point B_2 is slightly before that of maximum pressure as shown by A, and corresponds to the sudden melting of the wire. With this mixture fired at the centre the wire was always found to melt.

The comparison of temperature at the centre of the vessel with mean temperature as determined by pressure therefore ceases about 0·025 second before maximum pressure is attained.

Consider now the curve D: it is not deflected till the point D_1 is reached, and here the temperature begins slowly to rise; at D_2 there is a sudden rise, just 0·01 second before the centre wire melts, and 0·035 second before maximum pressure is attained. The temperature attained by this rise is maintained for nearly one second and then slowly falls.

By taking the mean of successive maxima and minima, Professor Hopkinson has prepared the following table showing the temperature of the thermometer, B at the centre of the vessel reckoning time from the point B_1 on the diagram (fig. 67) till the point of melting the platinum.

EXPLOSION IN A CLOSED VESSEL. (*Hopkinson*)

Initial pressure, 14·7 lbs. per sq. in. abs.
Initial temperature, 20° C.

Time from start of ignition-point B[1]	Temperature (° C.) by platinum thermometer at centre of vessel	Time from start of ignition-point B[1]	Temperature (° C.) by platinum thermometer at centre of vessel
0·008	560	0·107	1260
0·024	995	0·123	1275
0·041	1135	0·140	1275
0·057	1165	0·173	1400
0·074	1165	0·26	1710 wire melts
0·09	1225	—	—

The maximum temperature shown on the curve D, Professor Hopkinson states, is about 1250° C., and occurs very nearly at the moment of maximum pressure. This estimate he says, however, may be a good deal wrong, because the diagram being on a smaller scale cannot be as accurately measured as B.

As the result of a considerable number of experiments with this mixture always fired at the centre Professor Hopkinson considers that the distribution of temperature at the moment of maximum pressure is roughly as follows :

Mean temperature (inferred from pressure)	1600° C.
Temperature at centre of vessel, thermometer B . . .	1900
Temperature 10 cm. (4 in.) within wall, thermometer C .	1700
Temperature 1 cm. (0·4 in.) from wall at end, thermometer D	1100 to 1300
Temperature 1 cm. (0·4 in.) within wall at side . . .	850

At the thermometers B, C, and D (he states) the gases can have lost but little heat at this period, and the temperature differences are mainly due to the different treatment of the gas at different places. At B it has been burned nearly at atmospheric pressure, and compressed after burning to about 6½ atmospheres absolute; at D it has been compressed to about 6 atmospheres as in a gas engine, and then ignited without any subsequent compression; at the point 1 cm. within the wall at the side much heat has been lost, as this is the first point reached by the flame; the gas here is ignited when the pressure is about 2 atmospheres; its temperature rises instantly to 1300° C., and at once begins to fall.

Half a second after maximum pressure the distribution is very different: convection has now had time to take effect.

The distribution of temperature is broadly as follows:

Mean temperature inferred from pressure	1100° C.
Mean temperature, exclusive of layer 1 cm. thick at walls, determined by long platinum wire from B to D . . .	1160
Temperature at centre of vessel, thermometer B .	1100 to 1200

Here the temperature differences are much smaller than at the moment of maximum pressure. The mass of gas during cooling may be described as a hot core in which the temperature is approximately uniform, varied accidentally by currents, surrounded by a thin layer wherein the temperature falls to the temperature of the walls. Professor Hopkinson calculates that if such layer were $\frac{1}{2}$ cm. thick, and if the fall of temperature were uniform, the mean temperature inferred from the pressure would fall short of that of the hot core by about the observed amount, that is, 60° C.

Assuming both air and gas to be saturated with moisture, Professor Hopkinson calculates the composition of the products of combustion from the mixture used in the experiment—9 of air to 1 of gas—to be as follows:

Carbonic acid, CO_2	= 5·78	per cent. by volume
Water, H_2O	= 15·80	,, ,,
Nitrogen and oxygen, N and O	= 78·5	,, ,,
	100·08	per cent. by volume.

This assumes that the CO_2 and H O occupy their proper molecular volumes at 20° C.

The following are the most important of Professor Hopkinson's deductions with regard to this mixture:

Velocity of Spread of Flame.—This is found to be roughly 150 cm. (59 ins.) per second between the thermometers B and D. Mallard and Le Chatelier's value for a mixture containing 17 per cent. of coal gas in a tube 2 to 3 cm. diameter was 125 cm. per second.

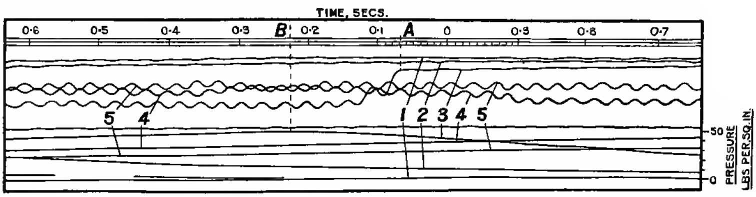
Period of Explosion when Flame fills Vessel entirely.—The explosion vessel is entirely filled with flame when the pressure reaches 70 lbs. per sq. in., although the maximum pressure attained later is 82 lbs. per sq. in. above atmosphere.

Maximum pressure is attained in $\frac{1}{30}$ of a second after complete filling of the vessel by flame.

Maximum Temperature at Centre of Vessel partly due to Compression after Ignition.—The temperature at the centre of the vessel rises rapidly after ignition to about 1225° C., which it reaches within about $\frac{1}{20}$ second. This temperature remains nearly stationary during the early part of the spread of the flame; the pressure during this time

remains nearly constant. After this the mass of gas near the centre, at about 1200° C., is compressed to 6·5 atmospheres, nearly adiabatically to a temperature of about 1900° C. Assuming a loss by radiation from the flame of about 15 per cent., then the specific heat of the products of combustion between 1200° and 1900° C. is 1·3 times that of air, and the average value of γ (ratio of specific heats) is 1·25 between 1200° and 1900° C.

Temperature Differences would exist in a Gaseous Explosion ignited in a perfectly non-conducting Vessel.—Hopkinson's experiments distinctly prove that temperature differences would exist after complete inflammation, even in an entirely non-conducting vessel. The highest temperature would be reached at the point of origin of the ignition due to the compression of the gases first heated by explosion to about 1200° C., and then compressed by the compression from the walls inward as the mixture near the walls inflames.



The successive portions of the pressure and temperature curves are numbered in the order of the corresponding revolutions of the drum

FIG. 68.—Explosion and Cooling Curves of gas and air mixture (1 gas and 12 air) at atmospheric pressure, with temperature recorded by platinum resistance thermometer. (*Hopkinson*)

This is the most novel point found by Hopkinson's investigations.

It was pointed out by the present writer in 1886 that a gaseous explosion in a cooling vessel consists of a hot core and a cool zone in contact with the walls; but Hopkinson's point was not discovered, although, now that it has been proved, it is obvious that the phenomenon should have been capable of prediction.

Explosion of Weak Mixture, 1 *Gas,* 12 *Air.*—With 12 volumes of air to 1 of gas the maximum pressure of the explosion is about 50 lbs. per sq. in. above atmosphere, and it is attained in 2·5 seconds after the passage of the spark.

Fig. 68 shows a copy of the photographed diagram produced with its pressure line and one temperature line.

A large number of diagrams were taken with the platinum thermometer in all sorts of positions, but in this case only one wire was used, and it was placed 15 cm. (5·9 ins.) from the spark and vertically below it. At first the temperature in the wire rises very slowly.

More than 2 seconds after ignition (at A) it is only about 210° C., and this heating is almost entirely due to adiabatic compression. The flame reaches the thermometer at the point A, and the temperature rises in $\frac{1}{10}$ second to 1300° C., at which point (B) the pressure has attained its maximum value, about $2\frac{1}{2}$ seconds after the spark has passed. The temperature is steady for a while and then falls, but there is no perceptible rise after the pressure begins to fall. Hopkinson states that in a large number of trials he was unable to discover any

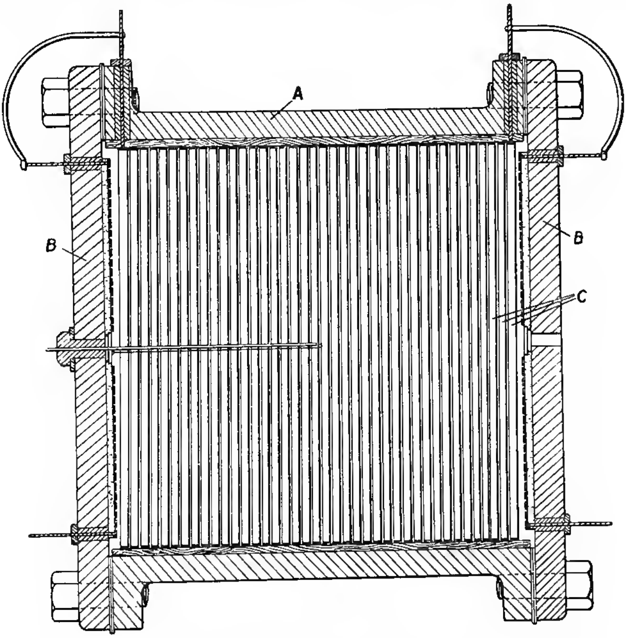


FIG. 69.—Recording Calorimeter for Explosions. (*Hopkinson*)

point at which inflammation occurred later than in this diagram, so that it is probable that all the gas is ignited at the time of maximum pressure.

Hopkinson's Experiments with a New Recording Calorimeter for Explosions

If the heat-flow from a mass of hot gases contained within a cylinder could be directly measured as the heat passes from the gas into the walls, it would be possible to avoid many difficulties in the determination of the questions of varying specific heat, continued combustion, law of heat loss, and so forth. The apparatus would have

the advantage of independence of specific heat determinations by combustion and other methods. Professor Hopkinson has attacked the difficult problem with a most ingenious apparatus, which he describes in the following way:

'It consists essentially in lining the explosion vessel as completely as possible with a continuous piece of copper strip and recording the rise of resistance of the copper strip during the progress of the explosion and the subsequent cooling. Knowing the temperature of the copper and its capacity for heat, the heat that has flowed into it from the gas may be calculated from the resistance.'

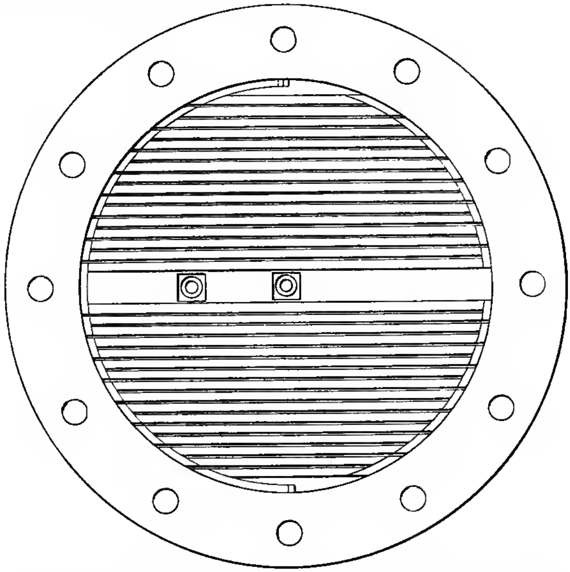


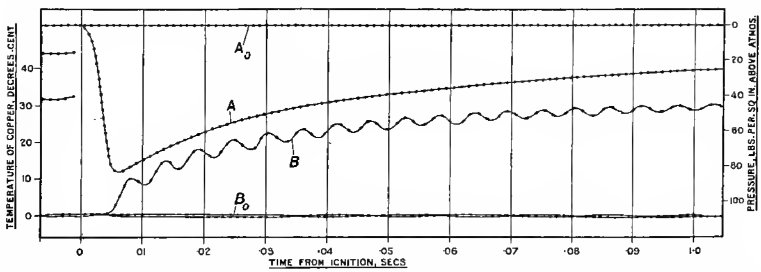
FIG. 70.—Recording Calorimeter for Explosions. (*Hopkinson*)

The explosion vessel is shown in longitudinal section in fig. 69, and the end of the vessel in elevation at fig. 70.

'It consists of a cast-iron cylinder A, 1 foot in diameter and 1 foot in length, on to which are bolted two end-plates B. The cylinder was first completely lined with wood $\frac{1}{4}$-in. thick and the end-plates covered with pieces of cork.

Thirty-nine turns of copper strip C of the quality used for electric lighting purposes, and of a high degree of purity, were then wound on the inside of the curved portion, a clearance of about $\frac{1}{20}$ of an inch being left between successive turns. The strip was $\frac{1}{4}$-in. wide by $\frac{1}{25}$-in. thick. The ends of this piece of strip were brought to terminals outside the vessel. The end-plates were similarly covered

with parallel pieces of copper strip of the same dimensions, as shown in fig. 70, the ends being brazed to connecting pieces. The strips on the ends were electrically connected outside the vessel to the strip in the cylindrical part. The whole when put together formed an explosion vessel having a capacity of about 0·684 cub. ft., which (except for the uncovered portions on the ends where the cocks, &c., came through) was completely lined with an electrically continuous length of copper having an approximately uniform section of $\frac{1}{100}$ of a square inch. For recording the pressure I used an optical indicator, consisting of an iron piston which was forced by the pressure against a piece of straight spring held at the ends. The displacement of the spring tilted a mirror about a fulcrum, and the mirror cast an image of a fine hole illuminated by an arc lamp on to a photographic film carried on a revolving drum. This indicator was repeatedly calibrated by dead



Pressure before ignition (atmospheric), 14·6 lbs. per sq. in. Maximum pressure, 90 lbs. per sq. in. above atmosphere. Temperature before ignition, 15° C. Maximum temperature of explosion, 1760 ° C.

FIG. 71.—Explosion and Cooling of mixture (gas 1, air 6·88 vols.) with copper-strip calorimeter, curve showing heat flowing to walls during explosion and cooling. (*Hopkinson*)

weights, and I think its readings are to be trusted to within 1 per cent. of the maximum reached. The mixture was fired by an electric spark at the centre of the vessel, and it was at atmospheric pressure and temperature before firing. When the explosion takes place the copper strip is heated and its resistance rises, and since the current in it remains constant during the short time occupied by the cooling of the gas to ordinary temperatures, the potential at the terminals of the strip rises by an amount proportional to the increase of resistance or to the increase of temperature. Since the potential at the terminals of the resistance remains constant, except for the small disturbance due to the passage of the galvanometer current, the galvanometer deflection from the reading just before the explosion will be proportional to the rise of potential between the terminals of the strip or to the rise of temperature. The mirror of the galvanometer reflected on to the moving film an image of the same small hole as

was used for recording the change of pressure, and a simultaneous record was thus obtained of the change of temperature of the strip and of the pressure in the vessel.'

Fig. 71 shows a record of the change of pressure due to explosion and cooling, and the change of temperature of the copper strip which forms the calorimeter. Curve A is the pressure measured downwards from the atmospheric line A_0 and curve B is the galvanometer deflection measured upwards from the zero line B_0.

The galvanometer is thrown into oscillation by the rapidity of the first heat addition from the explosion. An arc using an alternating current was the source of light for illuminating the aperture; hence white dots were shown in the photograph, and these are useful to identify corresponding points on the two curves.

The following are the particulars of the experiment which gave the record, fig. 71.

Mixture used, 1 *gas,* 6·88 *air*

Vessel, 0·684 *cub. ft. capacity*

Cambridge coal gas	0·082 cub. ft. =	12·7 per cent.
Air (including some water vapour) . .	0·565 =	87·3
Total	0·647 =	100·0

These volumes are at standard temperature and pressure 0° C and 760 mm.

Calorific value of the gas by Boys' calorimeter at 0° C. and 760 mm. per cub. ft.

Higher value 670 British thermal units or 372 Centigrade heat units = 170,000 gramme calories per cub. ft.

Pressure before explosion, 753 mm. = 14·6 lbs. per sq. in.

Temperature before explosion, 15° C. = 288° C. absolute.

Products of Combustion

Carbonic acid	CO_2 =	0·046 cub. ft. =	7·4 per cent.
Steam	H_2O =	0·118 =	18·9
Nitrogen and oxygen		0·462 =	73·7
Volume of products assumed to be gaseous at 0° and 760° mm. .		0·624 =	100·0

Holborn and Austin's determinations of specific heat of carbonic acid, steam, nitrogen, and oxygen have been taken.

Carbonic acid, CO_2 = 10·7 calories per standard cub. ft., mean value between 15° C. and 545° C.

Steam under same condition, 8·4 calories.

Nitrogen and oxygen, 6·3 calories.

These are specific heats at constant volume, all in gramme calories.

Allowing for a contraction of 3 per cent. for the combustion of

Cambridge gas (1 gas and 7 air) on the original photographic record, 1 mm. on the pressure diagram corresponds to a temperature rise of 36°·6 C., and on the galvanometer curve B a rise of 1 mm. is equivalent to a mean temperature rise of the strip of 0°·83 C., or a heat quantity of 222 gramme calories.

Fig. 71 is reduced from the original film, but the temperature and pressure are marked.

'Of the heat which passes into the copper, some part is lost to the wooden backing behind it, and it is the balance only which is directly measured in the diagram. The percentage of heat so leaking out is a correction which increases from less than 1 per cent. 0·1 second after firing up to about 20 per cent. 1 second after firing. In order to determine the amount of this correction, resource was had to a method of electrical heating. . . .

'In addition to the heat which passes into the copper and, *viâ* the copper, into the backing behind it, heat also goes into those parts of the walls which are not covered.'

Professor Hopkinson estimates that the whole heat which the gas has lost exceeds that which has gone into the copper in the ratio $\frac{4000}{3780}$, or 6 per cent.

To test the accuracy of the new calorimeter, Professor Hopkinson considers the point on the cooling curve A, fig. 71, one second after ignition. Here the gaseous mass has fallen to a mean temperature of 545° C., and the corresponding point on the curve B shows the heat actually in the copper strip at the moment to be 7,850 calories. The heat passed through to the backing Hopkinson calculates 1,570 calories, or 20 per cent. of the heat in the copper. The total heat in and passed through the copper is thus 9,420 calories. Multiplying this by 1·06 gives 10,000 calories as the total heat-flow from the gaseous contents to the walls from the moment of ignition to the moment when cooling has reduced the gases to 545° C. Comparing this determination with the total heat in the gas and the amount remaining when cooling from 545° to 15° C. by Holborn and Austin's figures, already given, he gets

Total heat of combustion of 0·647 cub. ft. of the mixture	14,000 calories.
Heat of products evolved on cooling from 545° to 15° C.	3,807 ,,
	10,193 gramme calories.

That is, if Holborn and Austin's and other constants be accurate, the gas contents, in burning to maximum temperature and cooling from it to 545° C., should evolve, say, 10,200 calories. Hopkinson's instrument

shows 10,000, a satisfactory correspondence. Calculating, however, to 0·5 second from firing, the agreement is not so good. The temperature reached is 840° C., and the total heat lost to the walls, as shown by the instrument, is 7,980 calories; but from Holborn and Austin's figures it should be 8,820 calories, so that the instrument appears to register about 10 per cent. too low.

Hopkinson accounts for this partly by possible error in Holborn and Austin's values, but finally comes to the conclusion that at half a second after firing part of the gas is still unburned.

Fig. 72 is an interesting curve, taken from Professor Hopkinson's paper, which shows the heat loss from the gases to the copper strip in calories per square centimetre at different times from the moment of firing. The mean gas temperatures at the different times have

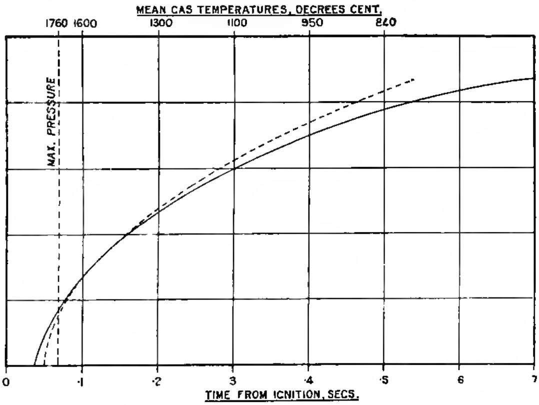


FIG. 72.—Curve of Heat Loss per square cm. at different times from ignition, from Experiment, Fig. 71. (*Hopkinson*)

been added. The maximum mean temperature of this explosion was 1760° C., and it fell to 840° C. 0·5 second after firing. An examination of this curve shows equal heat loss per degree fall from 1760° to 840° C.

With regard to the rate of loss Professor Hopkinson states:

'The loss of heat begins about $\frac{1}{20}$ second after ignition, when the flame first comes into contact with the copper. At first the loss goes on at a very great rate, and by the time maximum pressure is reached (when the flame is in contact with the whole surface of the strip and losing heat to every part of it), about 1,700 calories, or 12 per cent. of the gross heating value of the gas, has passed into the walls. The rate of loss of heat at this point is about 10 calories per second per square centimetre, and the mean gas temperature is 1760° C.

At 0·2 second from ignition the rate of heat loss is about $3\frac{1}{2}$ calories per second, and the mean gas temperature is 1300° C. The mean temperature is reduced in the ratio 0·74 between these two points, the product of mean temperature and pressure is reduced in the ratio 0·55, but the rate of loss of heat at 0·2 second is only one-third of what it is at maximum pressure.'

CHAPTER VII

EXPLOSION AND COOLING IN A CLOSED VESSEL—DISCUSSION OF DATA DEDUCIBLE

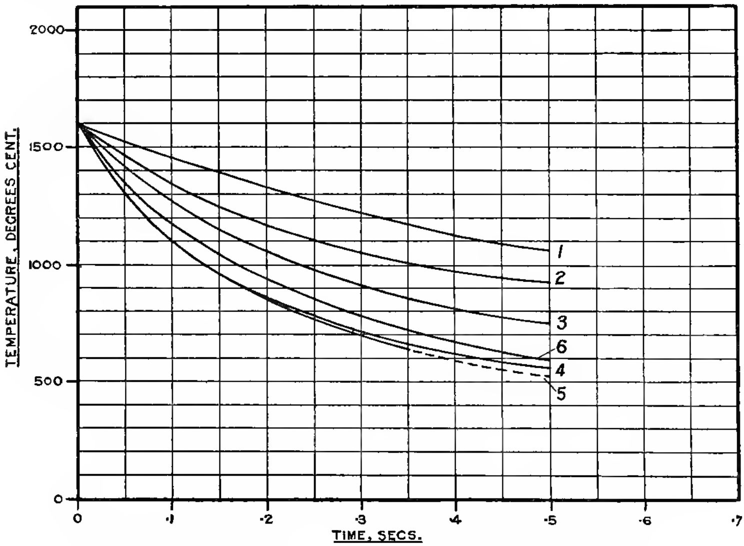
THE experiments made upon explosions of mixtures of coal gas and air have been given in some detail in the previous chapter, because it is desirable for the engineer to understand what has been done and the difficulties which have been met by different investigators. As will be seen, a considerable amount of experimental knowledge is now available for the purpose of developing the general principles underlying the use of gaseous explosions for producing motive power; but there remain many points of difficulty to be disposed of before it can be said that our knowledge of the working fluid is complete. Some things, however, have been definitely settled. Holborn and Austin's investigations have placed it beyond doubt that the specific heat of steam and carbonic acid increases considerably with increase of temperature, and that a small increase occurs with oxygen and nitrogen. Nernst's investigations have proved that the dissociation of steam and carbonic acid at about 2000° C. is unexpectedly small.

Using the new specific heat values, there is still a considerable discrepancy between calculated and observed temperatures, so that the question of continued combustion has still to be considered. It is not desirable to discuss these matters at this stage, because happily many general deductions may be drawn without relying on disputable theoretical points. It is proposed, then, to discuss the experiments from the point of view of the engineer-designer desirous of having some knowledge of the working properties of his working fluid. As the first thing required is an accurate conception of the losses to the enclosing walls, this will now be considered for mixtures of similar composition ignited at an initial pressure of one atmosphere.

Explosion and Cooling. Initial Pressure Atmospheric.—Broadly, the experiments may be taken as establishing that when mixtures of similar composition are ignited either in large or small vessels at an initial pressure atmospheric, the maximum pressures attained are the same except where the ignition is slow and cooling loss is experienced

during the explosion period. The smallest vessel contained 0·150 cub. ft. and the largest 6·2 cub. ft.

It may be that there is a small difference masked by this and mixing difficulties, but no such difference has so far been established. Although explosion pressures may be considered to be practically independent of dimensions of vessel, the rate of cooling varies greatly with dimensions.



1. Hopkinson's Large Vessel
2. Bairstow and Alexander
3. Hopkinson's Small Vessel
4. Clerk's Vessel, 1884, 1885
5. Boston Experiments
6. Clerk's Vessel, 1900

FIG. 73.—Cooling after Explosion in Closed Vessels of different dimensions from temperature 1600° C. for 0·5 second

Cooling from the same Mean Temperature in Vessels of Different Dimensions.—To enable the rate of cooling to be compared, cooling curves given by six vessels have been drawn; they are:

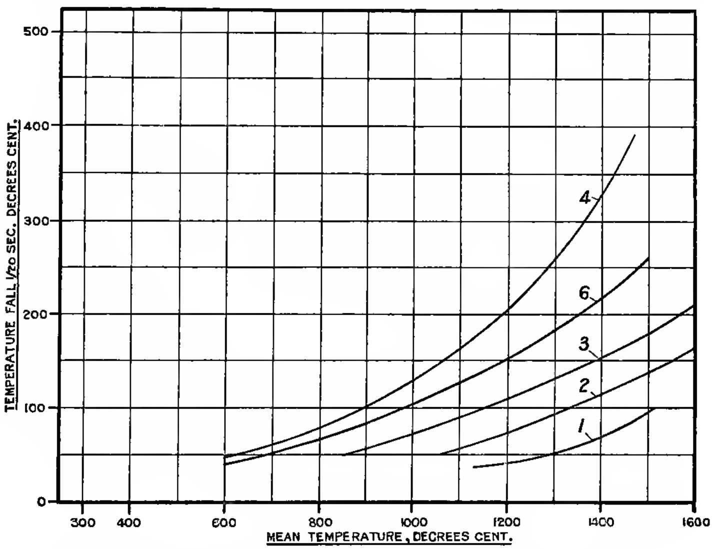
(1) Hopkinson's large vessel	6·2 cub. ft. capacity.
(2) Bairstow and Alexander's vessel .	0·82 ,, ,,
(3) Hopkinson's small vessel	0·684 ,, ,,
(4) Clerk's vessel	0·183 ,, ,,
(5) Boston experiments vessel . . .	0·180 ,, ,,
(6) Clerk's vessel, 1900	0·150 ,, ,,

The curves start from the same maximum 1600° C., at which point time is made zero; the curve is carried to 0·5 second, so that the cooling

curves start at zero of time at 1600° C. and fall to various temperatures one half-second afterwards.

Fig. 73 is a diagram showing these six cooling curves.

The gases in vessel (1) have cooled in 0·5 second through about 530° C., while those in vessel (4) have fallen through approximately 1030° C. in the same time. Compare the temperature fall in 0·2 sec. (one-fifth of a second), as has been done in the case of the author's earlier experiments, so as to use a period of the order of the complete working stroke of an ordinary gas engine. At the end of 0·2 second the temperature fall in vessel (1) is about 270° C.; in vessels (4) and (5) about



1. Hopkinson's Large Vessel
2. Bairstow and Alexander
3. Hopkinson's Small Vessel
4. Clerk's Vessel, 1884, 1885
6. Clerk's Vessel, 1900

FIG. 74.—Cooling after Explosion in Closed Vessels of different dimensions; curves showing fall of temperature in $\frac{1}{20}$th second for equal mean temperatures

740° C. It is obvious, then, that the temperature is falling much more slowly in the large vessel than the small one. Owing to the rapid fall of temperature in the small vessel as compared with the large one, the mean temperature during the periods compared is lower in the small vessel. To compare heat loss more accurately it is necessary to measure the temperature fall in each vessel for a much shorter period, and to so arrange matters that temperature falls are compared not only for equal times but also for equal mean temperatures.

Fig. 74 has been drawn to make this comparison possible. The

fall of temperature in $\frac{1}{20}$ second has been plotted against the mean temperature in that period of time.

In the following table the fall of temperature in the different vessels for $\frac{1}{20}$ second for mean temperatures 1450°, 1400°, 1300° and 1150° C. have been taken from these curves.

FALL OF TEMPERATURE IN DIFFERENT VESSELS FOR $\frac{1}{20}$ SECOND AT DIFFERING MEAN TEMPERATURES

Initial pressure . . . . Atmospheric

Vessel		Temperature falls in $\frac{1}{20}$ sec. from mean temperature			
No.	Capacity	1450° C. mean temp.	1400° C. mean temp.	1300° C. mean temp.	1150° C. mean temp.
	cub. ft.				
1	6·2	80° C.	68° C.	52° C.	38° C.
2	0·82	127° C.	114° C.	93° C.	65° C.
3	0·684	166° C.	153° C.	131° C.	100° C.
4	0·183	372° C.	327° C.	257° C.	182° C.
5	0·180	372° C.	327° C.	257° C.	182° C.
6	0·150	238° C.	216° C.	184° C.	138° C.

From this table it is evident that for equal mean temperatures the temperature fall is less in the large vessels than in the small.

It would be expected that, if a mass of gas be in contact with a cold surface at a given temperature and a given density, it would lose heat to that cold surface at the same rate whether it be contained in a large or small vessel. This, however, assumes that the film in contact with the walls is of the same temperature in all sizes of vessels. If this be true, then the heat loss per square foot expressed for hot mixtures of the same chemical composition should be the same for identical temperatures and times. It is therefore necessary to compare the heat losses above given as to quantity of heat. In order to avoid using specific heat values or continued combustion values, it is desirable to make the comparison in terms of temperature fall.

As 1° C. temperature fall represents a larger absolute loss in a large as compared with a small vessel, the author will adopt as a unit of heat quantity that amount of heat which is given out by the fall of 1 cub. ft. of the glowing gas at constant volume by 1° C.

The gas to be measured in the first instance at atmospheric pressure and a standard temperature.

Call this heat quantity *one cubic foot degree*. The capacity of vessel (1) is 6·2 cub. ft., so that the heat quantity lost by cooling for $\frac{1}{20}$ of a second at a temperature of 1450° C. is $80 \times 6·2 = 496$ cub. ft. degrees. With vessel (6) $238 \times 0·150 = 35·7$ cub. ft. degrees, the total heat

lost in the large vessel during the time was 496 cub. ft. degrees, and in the small 35·7 cub. ft. degrees.

The internal surfaces of the different vessels are approximately :

Number	Capacity	Internal surface
	cub. ft.	sq. ft.
1	6·2	17·3
2	0·82	5·02
3	0·684	4·33
4	0·183	1·79
5	0·180	1·79
6	0·150	1·60

If now the heat quantities be divided by the surface we get the number of cub. ft. degrees lost in $\frac{1}{20}$ second per square foot exposed. This has been done for the mean temperatures below :

Cubic Foot Degrees lost in different Vessels for $\frac{1}{20}$ second, per Square Foot of Exposed Surface at differing Mean Temperatures

Initial pressure . . . . atmospheric

Vessel		Cub. ft. degrees			
No.	Capacity	1450° mean temp.	1400° mean temp.	1300° mean temp.	1150° mean temp.
1	6·2	28·6	24·4	18·6	13·6
2	0·82	20·7	18·6	15·2	10·7
3	0·684	26·2	24·2	20·7	15·8
4	0·183	38·0	33·4	26·3	18·6
5	0·180	37·4	32·9	25·8	18·3
6	0·150	22·3	20·2	17·2	12·9

In the experiments compared the mixtures of coal gas and air were practically the same, 1 gas, 9 of air; for vessels (1), (3) and (4) the mixtures (2), (5) and (6) were richer of about 1 gas to 7 of air. The Hopkinson experiments (1) and (3) were made with the same gas at Cambridge, and they show very similar heat losses per square foot as measured in cub. ft. degrees. The Bairstow and Alexander (2) and Clerk 1900 (6) values seem low, and the Clerk and Boston values (4) and (5) high. There is reason to believe, however, that the early explosion experiments of Clerk and the Boston experiments were not so reliable as the later experiments of Clerk, Bairstow and Alexander, and Hopkinson; and if the results from vessels (4) and (5) be rejected it will be noted that the other heat-loss values closely approach each other. Further experiments must be made to eliminate the uncertainty as to continued combustion. Meantime useful deductions

may be drawn as to the effect of dimensions on temperature fall by using the values suggested.

Assume, for example, a series of five cubical vessels with the sides respectively 0·5, 1, 2, 3, and 4 ft. Their cubic contents will be respectively:

0·125, 1, 8, 27, and 64 cub. ft., and their surfaces will be 1·5, 6, 24, 54, and 96 sq. ft.

Assume each cube to be filled with a mass of glowing gas at a mean temperature during $\frac{1}{20}$ second of 1300° C. Take 20 cub. ft. degrees as the heat loss for each square foot, then the heat lost by the respective vessels will be 30, 120, 480, 1,080, and 1,920 cub. ft. degrees.

Dividing these numbers by the contents of the vessels we have as the temperature fall in each vessel 240°, 120°, 60°, 40°, and 30° C.

Assuming 0° C. to be the lowest available point for cooling the gas at 1300° C., then these temperature fall values in percentages of the total range give 18·4 per cent., 9·2 per cent., 4·6 per cent., 3·08 per cent., and 2·3 per cent.

Assume the exposure to 1300° C. to last $\frac{1}{5}$ second instead of $\frac{1}{20}$, then these numbers must be multiplied by four. Multiply them:

73·6 per cent., 36·8 per cent., 18·4 per cent., 12.32 per cent., and 9·2 per cent.

These are the relative losses incurred during the period of the stroke of an ordinary gas engine by these masses of gas.

From these figures it is evident that loss by cooling may be reduced indefinitely by increasing the dimensions of the engine, so that when volumes of flame, such as 64 cub. ft., are used in an engine, which is the order of dimensions of many large engines now, heat loss may be rendered practically negligible even with a non-compression engine.

Taking the mean temperature 1150° C., and assuming 15 cub. ft. degrees to be the temperature fall per square foot per $\frac{1}{20}$ second, we get the following numbers as the temperature falls in the respective vessels in $\frac{1}{20}$ second: 180°, 90°, 45°, 30°, and 22·5° C.

The percentages of 1150° C. are as follows: 15·7 per cent., 7·8 per cent., 3·9 per cent., 2·6 per cent., and 1·95 per cent.

Obviously a smaller proportion of the total heat present is lost by working at a mean temperature of 1150° C.; the fall of 150° C. in the mean temperature effects a considerable saving.

The temperature fall in a given time for any vessel can be readily calculated if the heat loss per square foot exposed surface be assumed the same for all vessels; let c = capacity in cubic feet; s = surface exposed in square feet, and t = cubic foot degrees heat loss per square foot for the particular given time, and x = temperature fall for the time in the vessel,

$$x = \frac{s\,t}{c}.$$

For a cubical vessel with a side of a feet, the capacity is a^3 and the surface exposed is $6a^2$, so that

$$x = \frac{6a^2 t}{a^3} = \frac{6t}{a}$$

and for a spherical vessel with a diameter of d feet the capacity is $0·5236d^3$ and the surface exposed is $3·1416d^2$, so that

$$x = \frac{3·1416\, d^2 t}{0·5236\, d^3} = \frac{3·1416\, t}{0·5236\, d} = \frac{6t}{d}$$

That is, for cubes and spheres the temperature fall for equal circumstances is inversely proportional to the side of the square or the diameter of the sphere. This is also true for similar cylinders and other vessels of varying dimensions. The temperature fall in similar engine cylinders of different diameters may be taken as inversely proportional to the diameter.

Explosion and Cooling. Initial Pressure above Atmosphere.—The experiments appear to indicate that for similar gaseous mixtures the maximum temperature attained upon explosion increases slightly with increase of initial pressure; but as the rate of ignition also appears to increase slightly with increasing initial pressure, it may be that the diminished heat loss during the explosion period is sufficient to account for the difference.

In the Royal College of Science experiments, for example, a mixture containing 1 volume of gas and 5·3 volumes of air ignited at an initial pressure of 44·1 lbs. per sq. in. gave a maximum temperature of 2400° C., while in Petavel's experiments a mixture of 1 gas and 5·7 volumes air, ignited at an initial pressure of 1,094 lbs. per sq. in. gave a maximum temperature of 2483° C.

Calculated on the same basis, with the same allowance for chemical contraction, Clerk's explosion experiments at atmospheric initial pressure give a maximum temperature of 2100° C.

It may therefore be taken for the present that higher initial pressures do not increase the maximum temperatures of explosion to any extent sufficient to introduce a material error, if the time of explosion be sufficiently small.

The Royal College of Science vessel was 0·82 cub. ft. capacity, while Petavel's vessel was a sphere of 0·0195 cub. ft. capacity, and maximum initial pressure on the former experiments was 55 lbs. absolute and in the latter over 1000 lbs. absolute; so that our knowledge extends over a wide range of capacity. It may be taken, therefore, that, given sufficiently rapid ignition, maximum temperature is little if at all affected by dimensions of vessel with such mixtures. The rate of cooling, however, varies greatly with dimensions and initial pressures.

Cooling from the same Mean Temperature in the same Vessel with different Initial Pressures.—The Royal College of Science experiments made by Messrs. Bairstow and Alexander are the only experiments of a sufficiently complete character to enable deductions to be drawn as to the changes in the cooling curves produced by varying initial pressures. From their pressure cooling curves (see fig. 65) the writer has prepared the temperature time cooling curves shown at fig. 75. These curves show the effect of varying density on cooling from a common temperature of about 1720° C.

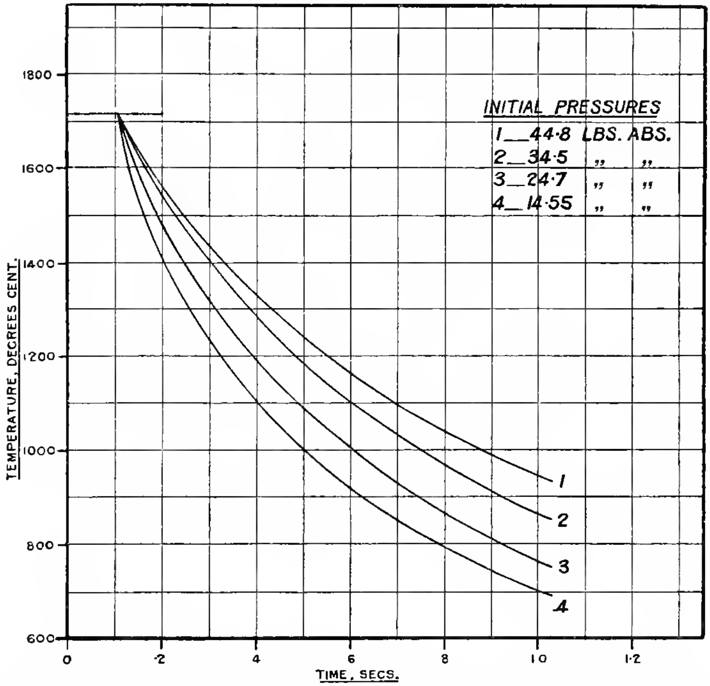


FIG. 75.—Cooling from the same mean temperature in the same explosion vessel with different initial pressures. After explosion of mixture containing 1 vol. gas to 5·95 vols. air.

The four curves are for initial pressures of 14·55 lbs., 24·7 lbs., 34·5 lbs., and 44·8 lbs. per sq. in. absolute ; all start from the common temperature of 1720° C.

It is evident from these curves that the higher the initial pressure the slower is the rate of temperature fall. Take the fall of temperature in 0·5 second, the highest density gives a fall of about 480°, while the lowest gives 715° C. This is shown in the table on the following page.

TEMPERATURE FALL IN THE SAME CLOSED VESSEL FOR DIFFERENT INITIAL PRESSURES

Common maximum temperature . . . . 1720° C.

Initial pressure	Temp. fall in 0·5 sec.
lbs. abs.	
44·8	480° C.
34·5	530° C.
24·7	630° C.
14·55	715° C.

As one-fifth of a second has been used as a period of the order of a gas engine stroke, fig. 76 has been prepared to show the fall of temperature from 1720° C. which occurs in $\frac{1}{5}$ second with the different densities. The temperature fall values are plotted against initial pressures.

TEMPERATURE FALL IN THE SAME CLOSED VESSEL FOR DIFFERENT INITIAL PRESSURES

Common maximum temperature . . . . 1720° C.

Initial pressure	Temp. fall in 0·2 sec.
lbs. per sq. in. abs.	
15	480° C.
39	335° C.
45	285° C.

As the mean temperatures vary considerably during 0·5 and 0·2 second, the curves shown at fig. 77 have been prepared, which give temperature falls in $\frac{1}{20}$ second for different mean temperatures with varying initial pressures. The initial pressures are marked upon the figure.

In the following table the fall of temperature in the same vessel ignited at different initial pressures is given for $\frac{1}{20}$ second for mean temperatures 1600°, 1500°, 1400°, 1300°, 1200°, 1100°, and 1000° C.

FALL OF TEMPERATURE IN THE SAME VESSEL FOR $\frac{1}{20}$ SEC. AT DIFFERENT MEAN TEMPERATURES AND DIFFERENT INITIAL PRESSURES

Vessel. Bairstow and Alexander. 0·82 *cub. ft. capacity*

Initial pressure	Temperature falls in $\frac{1}{20}$ second from mean temperatures						
	1600° C.	1500° C.	1400° C.	1300° C.	1200° C.	1100° C.	1000° C.
lbs. abs.							
44·8	74° C.	63° C.	54° C.	46° C.	39° C.	32° C.	25° C.
34·5	85° C.	72° C.	61° C.	52° C.	44° C.	37° C.	30° C.
24·7	139° C.	109° C.	85° C.	68° C.	54° C.	44° C.	34° C.
14·55	185° C.	140° C.	109° C.	87° C.	71° C.	59° C.	49° C.

Here it is evident that increase of initial pressure causes considerable diminution of temperature fall in $\frac{1}{20}$ second for equal mean temperatures. The increase from practically 1 to 3 atmospheres diminishes the temperature fall to less than half in the 1600° and 1500° C. values, and to half in the lower values.

The curves at fig. 78 show temperature fall for the different mean temperatures 1600°, 1500°, 1400°, 1200°, 1100° and 1000° C., plotted against initial pressures in atmospheres. The curves have been extended to 55 lbs. abs. to give an idea of the temperature falls in the region of such densities, which are now common in gas and petrol engine practice. Further experiments are necessary to obtain accurate

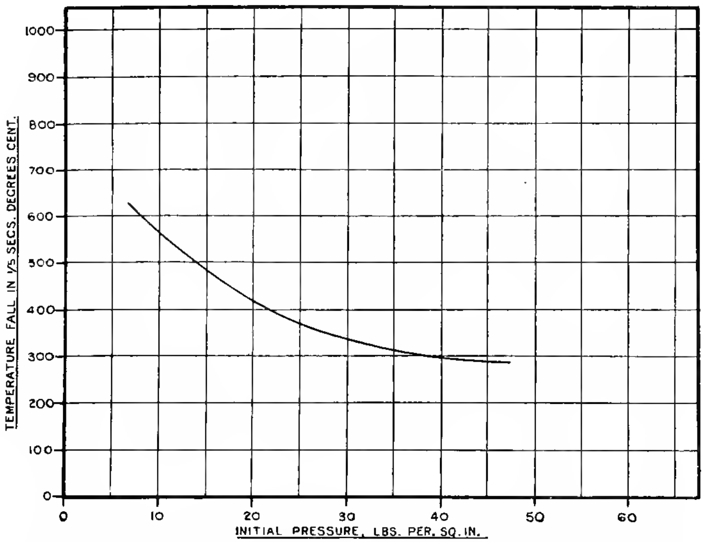


FIG. 76.—Temperature fall in $\frac{1}{5}$th sec. in the same closed vessel for different initial pressures. Common maximum temperature, 1720° C.

data, but these numbers materially aid us in giving some quantitative accuracy to our reasoning upon internal-combustion motor problems.

They show that an increase from 1 to 4 atmospheres density the temperature fall diminishes to less than one third at the mean temperatures 1600° and 1500°, and to less than one half at the lower temperatures.

The proportional heat loss incurred at higher initial pressures is therefore less than at lower pressures, so that it is economical in a gas engine to increase the mean density of the charge. The absolute

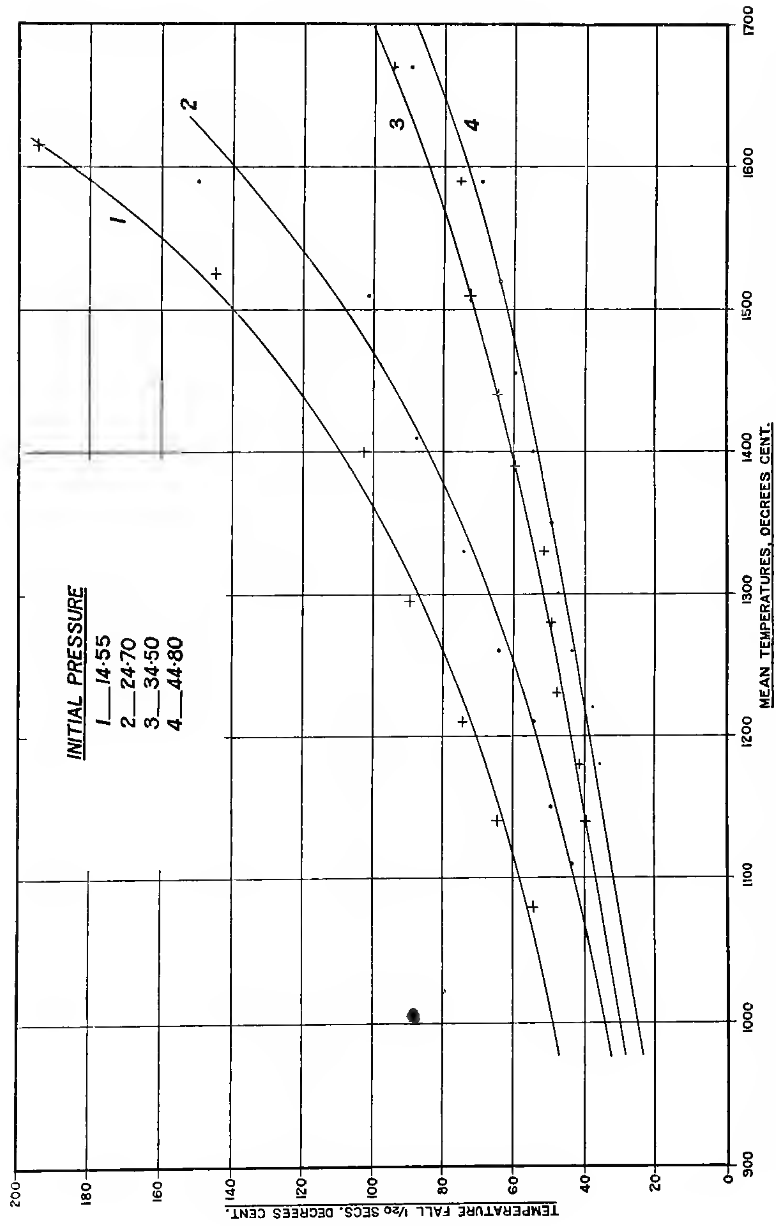


FIG. 77.—Temperature fall in $\frac{1}{20}$th sec. for different mean temperatures in the same closed vessel for different initial pressures

heat flow per square foot is greater, however, with higher pressures, as will be readily seen as follows :

Take the temperature fall at 44·8 lbs. (practically 3 atmospheres) at 1200° C. as 40° C. This would mean $40 \times 0{\cdot}82 = 32{\cdot}8$ cub. ft. degrees at atmosphere ; but there is three times the weight of gas cooling, so that the heat quantity is given by $32{\cdot}8 \times 3 = 98{\cdot}4$ cub. ft. degrees.

The surface of the vessel is 5·02 sq. ft. : $\frac{98{\cdot}4}{5{\cdot}02} = 19{\cdot}6$ cub. ft. degrees per square foot of surface per $\frac{1}{20}$ second. This is a heat flow which is about 50 per cent. greater than that which occurs with an initial pressure of one atmosphere for the same mean temperature.

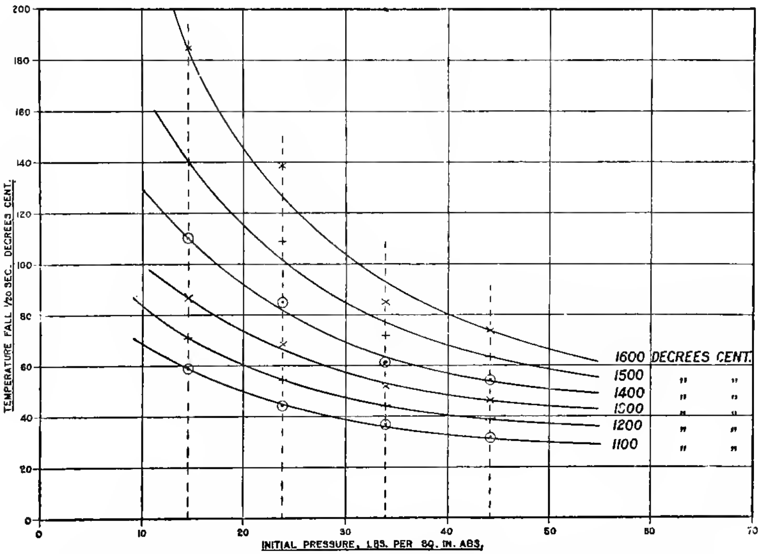


FIG. 78.—Temperature fall in terms of density for given mean temperatures

From this it follows that, although the percentage heat loss is diminished, a greater heat flow takes place per square foot of metal exposed, and therefore the metal is strained to a greater extent by unequal expansion.

If now the values given at p. 208 be multiplied by $\frac{0{\cdot}82}{5{\cdot}02} = 0{\cdot}163$, we get heat loss per square foot expressed in cub. ft. degrees at the particular initial pressures. The value of the cub. ft. degree will, however, be proportional to the initial pressure.

The table on p. 212 shows the values so obtained for 44·8 lbs. initial pressure and 14·55.

CUB. FT. DEGREES LOST IN THE SAME VESSEL FOR $\frac{1}{20}$ SECOND PER SQUARE FOOT OF EXPOSED SURFACE AT DIFFERING MEAN TEMPERATURES AND INITIAL PRESSURES

The value of the cub. ft. degree in this case being proportional to the initial pressure.

Capacity of vessel . . . . 0·82 cub. ft.
Surface exposed . . . . 5·02 sq. ft.

Initial pressures	Cub. ft. degrees loss from square foot per $\frac{1}{20}$ second						
	1600° C.	1500° C.	1400° C.	1300° C.	1200° C.	1100° C.	1000° C.
lbs.							
44·8	12·1	10·3	8·6	7·5	6·3	5·2	4·1
14·55	30·2	22·8	17·8	14·2	11·4	9·6	8·0

To realise the effect of increased density, make the same comparison as has been done at p. 205 for atmospheric initial pressure, using the five cubical vessels of 0·5, 1, 2, 3, and 4 ft. side, or 0·125, 1, 8, 27, and 64 cub. ft. capacity. Take the mean temperature 1300° C. and assume loss per $\frac{1}{20}$ second per square foot to be 10 cub. ft. degrees instead of 7·5, as given by the table; then the temperature fall in each vessel will be 120°, 60°, 30°, 20°, and 15° C.

Assuming 0° C. to be the lowest available point for cooling the gas at 1300° C., then these temperature fall values in percentages of the total range give—

9·2 per cent., 4·6 per cent., 2·3 per cent., 1·54 per cent., and 1·15 per cent.;

and for an exposure of $\frac{1}{5}$ of a second—

36·8 per cent., 18·4 per cent., 9·2 per cent., 6·16 per cent., and 4·6 per cent.

That is, the cooling loss may be reduced at 1300° C. to 4·6 per cent. of the total temperature range if the vessel be cubical and 64 cub. ft. capacity with the moderate compression ratio of 3.

Taking a mean temperature of 1100° C. and assuming the loss to be 5 cub. ft. degrees, the percentage losses for $\frac{1}{5}$ second are—

18·4 per cent., 9·2 per cent., 4·6 per cent., 3·08 per cent., and 2·6 per cent.

With an initial pressure of three atmospheres it is therefore possible to make proportional heat loss practically negligible in a vessel of 8 cub. ft. capacity, which corresponds to the cylinder capacity of an engine well within the range of present dimensions.

These experiments demonstrate that, so far as economy is con-

cerned, heat loss may be rendered negligible without having recourse to incandescent walls or non-conducting linings, which for a long time have formed the favourite device of the inventor who imagines that the greater part of the heat loss of an engine is through the enclosing walls.

It is also evident from the curves that the rate of reduction of heat loss diminishes with increased density, so that little increased heat saving would follow greatly increased density.

CHAPTER VIII

EXPLOSION AND COOLING IN A CYLINDER BEHIND A MOVING PISTON

In the preceding chapter it has been shown that fall of temperature per unit surface in a closed vessel depends upon—

Temperature difference;

Time of exposure;

Density of heated gas; and probably, to some extent,

Capacity of containing vessel when the vessel is small.

The larger vessels appear to give a smaller temperature fall per unit surface exposed in unit time, and this opens up many interesting but complex questions as to the effect of the history of the particular explosions upon the rate of cooling. When a vessel is very large and the duration of cooling is consequently long, it appears probable that a surface film of the metal attains a much higher temperature than is commonly supposed possible. Bairstow and Alexander consider that after a high temperature explosion of, say, 2000° C. maximum temperature a film upon the surface rises some hundreds of degrees and affects the subsequent rate of cooling. This conclusion they base upon Mallard and Le Chatelier's and Petavel's cooling curves; both show a discontinuity which they explain in this manner.

In the author's experiments, to be described in this chapter, he also finds that the interior surface of the cylinder and piston must attain relatively high temperatures compared with the temperatures of the water-jackets.

It cannot, therefore, be accepted without proof that the conditions of cooling are the same in an engine cylinder behind a moving piston as in a closed vessel of constant volume.

If it be assumed that the engine cylinder presents merely a succession of density, temperature, surface, and capacity changes, then the temperature falls to be expected can be determined from the foregoing data. To do this, however, it will be necessary to prepare curves of mean temperature, mean density, mean surface, for equal time-intervals in order to determine the temperature fall to be expected in a given cylinder; but at best this method assumes uniformity of conditions which are known to differ in many points. For example, the movement of the piston keeps the hot gases in motion, and whether

this motion be turbulent or not depends on many circumstances, such as shape of combustion chamber and absolute velocity of motion of the piston. Accordingly the author, some years ago, addressed himself to the problem of measuring the temperature fall due to cooling in the engine cylinder itself.

This is by no means a problem easy of solution, as the work done on the piston by the gases introduces complications difficult of elimination. No attempt had previously been made to determine a cooling curve for a cylinder having in it a moving piston.

The method of experiment adopted by the author was as follows :

The engine selected for the first experiments had a cylinder of 14 ins. diameter and 22 ins. stroke; the exhaust and inlet valve levers were supplied with longer pins than usual, so that the rollers mounted on these pins could be moved into or out of the range of the exhaust and inlet valve cams. When each roller was caused to slide to one end of its pin, the cam passed clear of it and the lever was not operated. When at the other end of the pin, the roller engaged with the cam and the lever operated in the usual way. A spring and trigger gear was so arranged that the rollers could be put out of range of the cams at any required instant. By this contrivance the engine could be run in its normal way in accordance with the Otto cycle either at a light or heavy load, and any given explosion could be selected for the purpose of the experiment by operating the trigger at the proper moment. It was thus possible to run the engine at its normal speed under the usual propelling explosions, and to select at any given moment any particular charge, move the rollers out of the range of the cams immediately the charge entered, and so obtain an explosion and expansion stroke in the usual manner, with the usual charge. When the exhaust period was approached, however, the exhaust valve remained shut, and accordingly the hot exhaust gases were retained in the cylinder and compressed by the return stroke of the piston into the combustion space at the end of the cylinder. The energy of the flywheel was sufficient to keep up the rotations of the engine, with but little fall in speed during the short period of observation. The piston was thus caused to move to and fro, alternately compressing and expanding the hot gases contained in the cylinder.

An indicator card taken of such an initial explosion and expansion and the subsequent series of compressions and expansions is given at fig. 79; *a b* is the ordinary compression line indicating the compression of the charge before explosion, *b c* is the usual explosion line, and *c* A the usual expansion line after explosion. At A, however, instead of the pressure falling to the atmosphere by the opening of the exhaust valve, as the exhaust valve remains closed no escape of the hot products of combustion is possible, and accordingly the return of the piston

produces the compression line A B; the next outward movement of the piston produces the expansion line B C, followed by the compression line C D; expansion line D E; compression line E F; expansion line F G; compression line G H; expansion line H I, and so on. In this diagram the successive compression and expansion lines have continued to be traced until the fall of pressure due to cooling brings the contents of the cylinder at the outer end of the stroke below atmospheric pressure, when the outer atmosphere opens the valves against the pressure of their springs, and so the experiment terminates. It will be observed that cooling is proceeding during the tracing of all these lines; had no cooling occurred or any particular expansion and compression stroke, the compression line would lie on the top of the expansion line. Con-

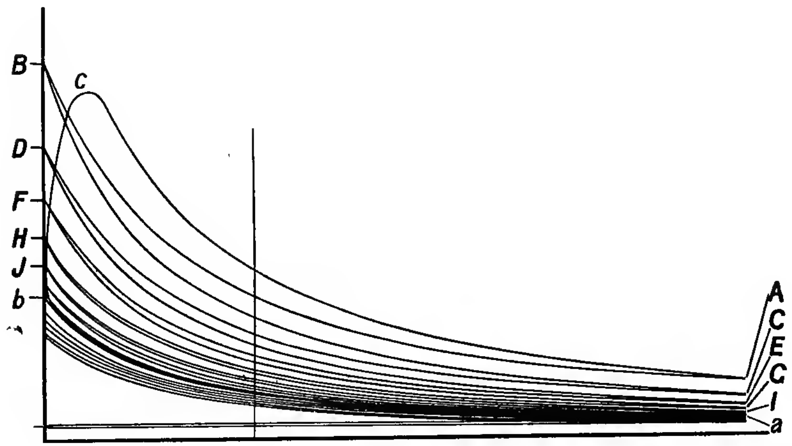


FIG. 79.—Clerk diagram of explosion and alternate compression and expansion of hot gases in engine cylinder

sider, for example, the moment represented by the point B when the piston is at the extreme inner end of its stroke; then the combustion space is filled with hot gases at a temperature and pressure corresponding to the point B; after a complete expansion and compression, one out- and one in-stroke of the engine, a complete revolution of the crank, the piston is again at its innermost position, and the whole of the gases are again contained in the combustion chamber at a pressure and temperature marked by the point D. This point D is lower than B, and as the weight of the gaseous contents has not changed it follows that the temperature at D is lower than at B. The points B, D, F, H, and J thus indicate the temperatures of the gases at the same volume at intervals of one revolution of the engine.

If the engine be running at 120 revolutions per minute, then the

temperatures represented by the points D, F, H, and J give the successive temperature falls suffered by the contents during successive revolutions, each lasting 0·5 second. In the same way the successive temperatures of the gaseous contents at the out ends of the stroke are given by the pressures at A, C, E, G, and I. Call the successive temperatures at the out end t_0, t_{01}, t_{02}, t_{03}, and t_{04} corresponding to the points A, C, E, G, and I, and the temperatures at the inner end of the stroke t_1, t_{11}, t_{12}, t_{13}, t_{14}, corresponding to the points B, D, F, H, and J. These temperature changes are partly due to heat loss from the gases to the cylinder walls and partly due to work done on the piston by the gases or by the piston on the gases. The points at both ends of the stroke when properly dealt with give curves of temperature fall due to heat loss to walls.

Fig. 80 shows a compression and expansion diagram without the explosion line.

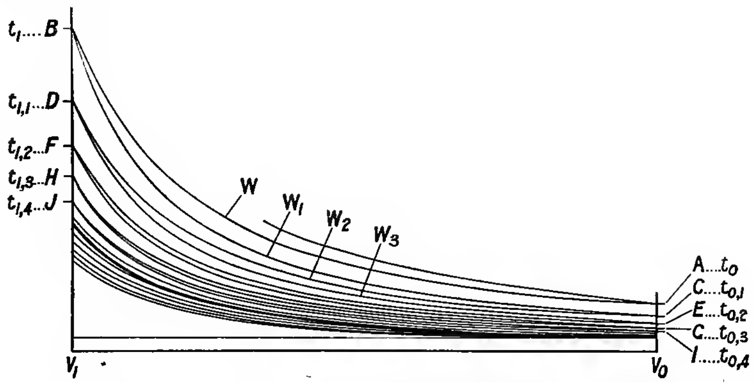


FIG. 80.—Clerk diagram of compression and expansion lines—explosion omitted

Consider first the series of points at the inner end of the stroke; obviously, if no cooling took place in the cylinder there would be no fall of temperature such as is shown between B and D. The fall between B and D is not entirely due to cooling, as will be seen when we consider what happens to the gases between the points B and D. When the engine piston expands the gases from B to C and compresses from C to D, a portion of the total heat has passed through the cylinder walls, but some work has also been done by the gases upon the piston. When the expansion BC takes place, the gases perform work on the piston equal to the area BC V_0 V_1; when the compression CD takes place, the piston performs work on the gases equal to the area CD V_1 V_0, so that more work has been performed by the gases on the piston than by the piston on the gases. Some work has therefore been done by the gases in the processes intervening between the points B and D; that is, part

of the temperature difference $t_1 - t_{11}$ is due to work done, it is not all due to heat lost through the cylinder walls. The difference between the two work-areas is BCD, so that the temperature fall $t_1 - t_{11}$ is due partly to heat loss to walls and partly to work done by the gas.

If the specific heat of the gaseous mixture at the temperatures between $t_1 - t_{11}$ be known, then the temperature fall due to the work area BCD can be calculated, and when deducted from the total temperature fall it gives the temperature fall due to heat flow through the cylinder walls. It is found that the specific heat value need only be approximately known, as the temperature fall equivalent of the work area BCD is small in comparison with the total temperature fall and little error is introduced by a considerable error in the specific heat value.

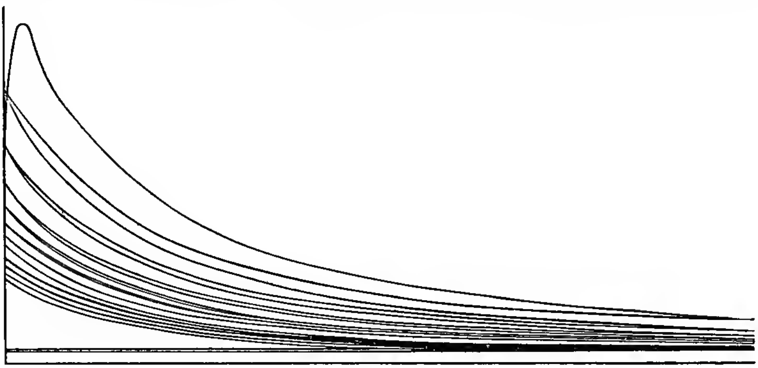


FIG. 81.—Clerk diagram from National Gas Engine of 14-in. cylinder by 22-in. stroke, with load on brake 50 HP. Jacket temperature, 71° C. Suction temperature, 95° C.

This method enables a true temperature fall curve to be drawn, showing the progressive fall of temperature incurred from revolution to revolution under the actual working conditions of the engine.

Proceeding in this way the author has determined the cooling curve or the engine referred to; it is an Otto cycle gas engne of 60 BHP by the National Gas Engine Co., Ltd., of Ashton-under-Lyne. The experiments were made with Ashton coal gas at the Company's works, Ashton.

Cooling Curve from 14-in. cylinder × 22-in. stroke National Gas Engine

Fig. 81 shows an indicator diagram taken by the author's method from this gas engine when working under a brake load of 50 HP at 160 revolutions per minute, water jacket at a temperature of about 71° C. A Richards-Casartelli indicator was used.

The temperature of the charge before compression was calculated at 95° C. The leading particulars are marked under the diagram.

In this diagram the successive temperatures at the points B, D, F, H, J are as follows:

	B	D	F	H	J
Temperature ° C. . .	1375	1035	815	665	560

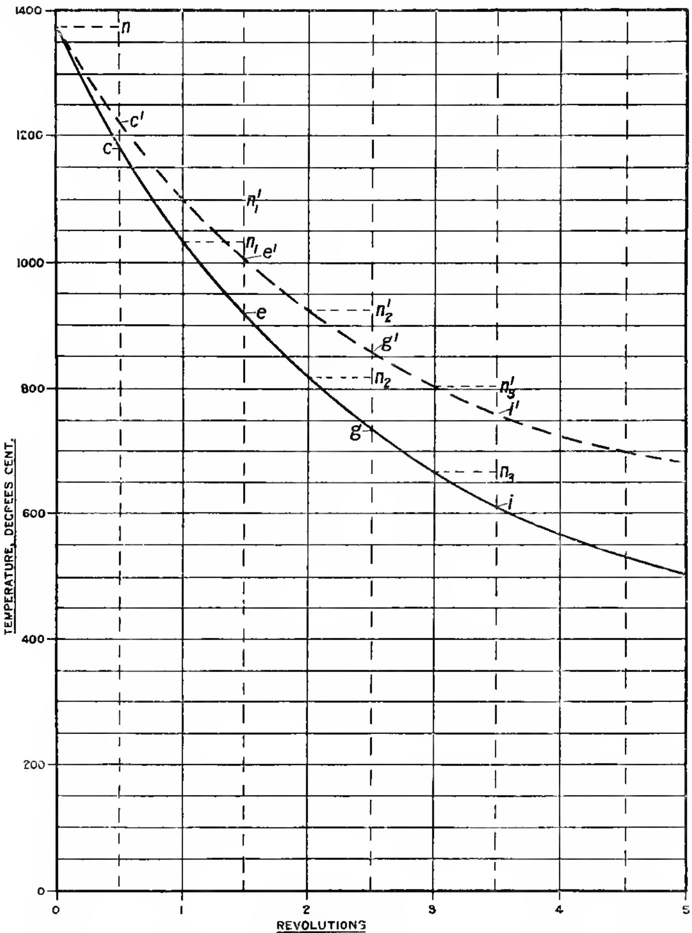


FIG. 82.—Temperature of contents of cylinder at inner end of stroke plotted against engine revolutions from fig. 81

That is, the temperature of the gases at the end of the first compression of the hot products of combustion was 1375° C.; the first complete revolution *after* found it fallen to 1035°; the second to 815°; the third to 665°, and the fourth to 560°.

Fig. 82 shows temperatures at these points plotted against complete revolutions as a full line.

Correcting for the work done upon the piston by the gases, we get the dotted line which gives the correct fall of temperature for each revolution due to heat loss through the enclosing walls.

The corrected temperature falls in these four revolutions are :

Revolution	1	2	3	4
Temperature falls in °C. . .	275	180	120	75

In the first revolution the gases have been expanded from B to C and compressed from C to D, so that the exposure of the contents to the relatively cold walls during one complete expansion stroke and one complete compression stroke at the particular mean temperature, in

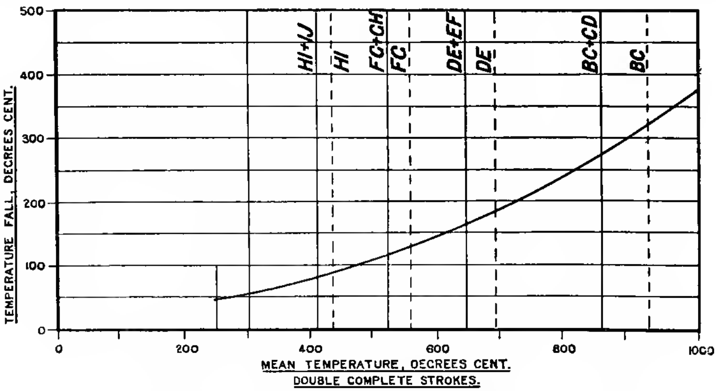


Fig. 83.—Temperature fall for double strokes plotted against mean temperatures during double strokes for diagram, fig. 81

time, has resulted in a heat flow to the walls, causing a temperature drop of 275° C. The second, third, and fourth strokes, although they occupy the same time and expose the same surface, yet expose the gases at a lower and lower mean temperature, so that the heat loss, and therefore temperature fall, becomes less and less as the gases approximate more and more to the temperature of the enclosing walls. To reason on these values it is necessary to plot the respective temperature falls against the mean temperatures existing during the double stroke in which the loss is incurred. This has been done at fig. 83, where the mean temperatures of exposure during complete double strokes is plotted against the temperature falls due to such exposure.

The full vertical lines marked on the figure indicate the mean temperature of the pairs of expansion and compression lines BC + CD,

DE + EF, FG + GH, and HI + IJ, while the dotted vertical lines represent the mean temperatures of the single expansion strokes BC, DE, FG, and HI.

The temperature falls given are those incurred in a double stroke at the particular mean temperature, whatever it may be, so that if the temperature fall in a single stroke is required, those temperature fall values must be divided by 2.

Consider the expansion lines. From the figure it is seen that the mean temperature of the expansion line BC is about 930° C., and the temperature fall due to heat flow from the gas in two strokes at this mean temperature is 320° C.—that is, in a single stroke the temperature fall would be 160° C. Now, as the engine is running at 160 revolutions

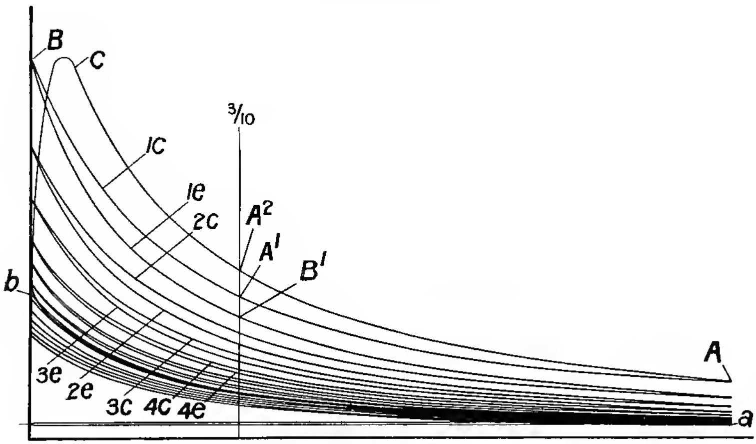


FIG. 84.—Clerk diagram arranged for calculating temperature fall at $\frac{3}{10}$ of the piston stroke

per minute, each revolution takes 0·375 second and each stroke 0·187 second, so that this is the temperature fall in this engine during a complete stroke at 930° C. mean temperature of 0·187 second duration. At 1000° C. mean temperature the temperature fall would be about 180° C., and at 1200° C. about 260° C. By this method it is possible to determine the law of temperature fall, and so to calculate the temperature fall which would truly represent the heat loss on the explosion expansion line. It is true that there is some turbulent motion in the gases during explosion and for some time after, but this method appears to give values which are very close to the truth, as will be seen later.

To get the heat loss temperature fall on the explosion and expansion line of an ordinary indicator diagram, it is only necessary to

calculate the mean temperature in time on the line, and the temperature fall may be deduced from such a diagram as fig. 83.

The examination of the expansion line BC has shown that its mean temperature is about 930° C., and its temperature fall due to heat flow through the cylinder walls 160° C. in the completed stroke. The question now arises, In what manner is this temperature fall incurred? Is it due to a uniform loss throughout the stroke, or does the temperature fall more rapidly at one part of the stroke than at another? We know from the previous chapter that increase of surface exposed increases heat flow and that increase of density also increases flow, but it would be difficult to predict what would occur throughout the stroke from the closed vessel experiment. The question, however, can be answered by the author's method of experiment.

The cooling curves, which have been already drawn (figs. 82 and 83), are those proper to complete revolutions. In order to answer this, it is necessary to deduce cooling curves for part of the stroke only, and for this purpose the first $\frac{3}{10}$ of the piston stroke has been chosen. Looking at fig. 84, it will be seen that a vertical line marked $\frac{3}{10}$ intersects the various compression and expansion lines; the partial compression lines have been marked respectively 1*c*, 2*c*, 3*c*, 4*c*, &c., and the partial expansion lines have been marked 1*e*, 2*e*, 3*e*, 4*e*, &c. Starting from the vertical $\frac{3}{10}$ line, at A′, the partial compression line 1*c* terminates at the point B, and the expansion line 1*e* passes from B to the $\frac{3}{10}$ line at the point B′.

In passing from A′ to B′ the hot gases have been compressed to B and expanded to B′, and the temperature has fallen from A′ to B′; the corresponding temperature fall, however, does not represent all the temperature fall due to heat flow, because the work done by the piston in the gas during the compression A′ B is greater than the work done on the piston by the gases during the expansion B to B′. This difference has disappeared as heat, and the temperature fall equivalent of the area A′ BB′ must be added to the apparent temperature fall A′—B′ to get the true value. By treating the other compression and expansion lines 2*c*, 2*e*; 3*c*, 3*e*; 4*c*, 4*e*, &c., the cooling curve shown at fig. 85 has been prepared as the mean of three experiments at full load cards under the conditions of fig. 81.

From the cooling curves for the whole stroke and $\frac{3}{10}$ stroke, it is now possible to consider the division of temperature fall between the first 0·3 and the last 0·7 of the stroke.

Consider first the expansion line BC.

From fig. 83 it will be found that during the whole stroke its mean temperature in time is 930° C., and the temperature fall during the stroke due to heat flow from the hot gases to the colder walls is 155° C.

From fig. 85 it will be found that the part B B′ of this expansion line—that is, 1*e*—has a mean temperature in time of 1190° C., and the temperature fall for this part is $\frac{185}{2} = 92\cdot5°$.

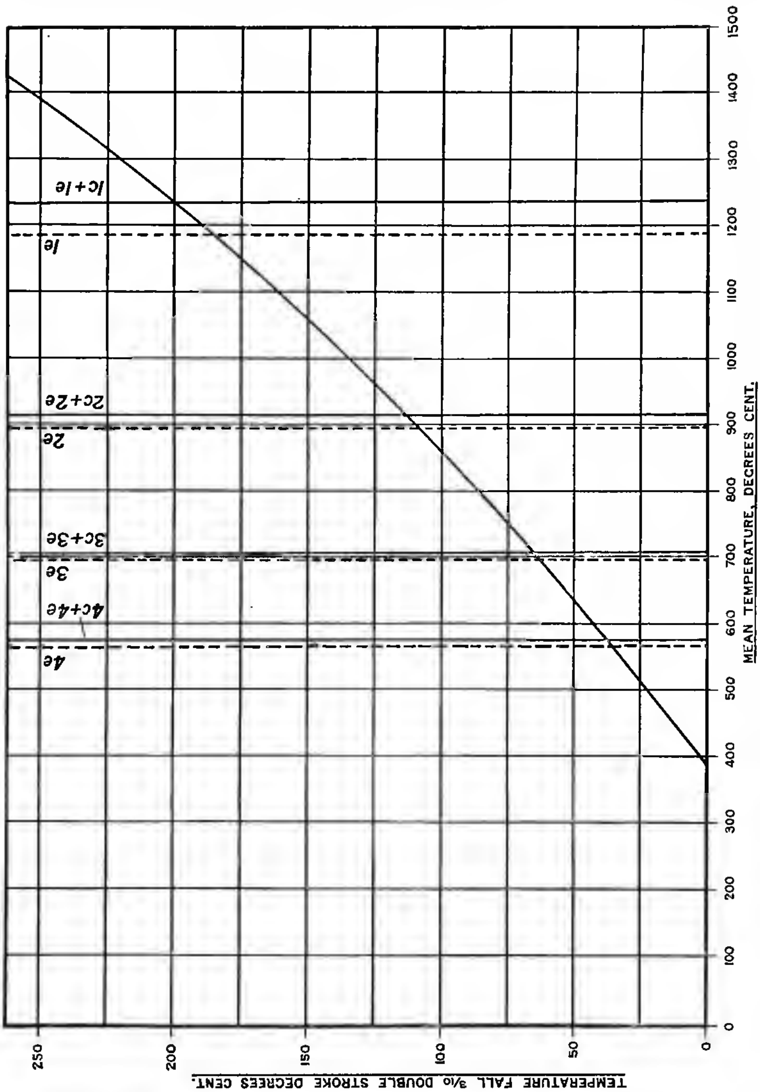


FIG. 85.—Temperature fall for $\frac{3}{10}$ double strokes plotted against mean temperature Mean from three full-load cards under conditions of fig. 81.

That is, while the temperature fall due to cooling on the whole stroke is 155° C., of that total fall 92°·5 C. is incurred in the first 0·3 of the piston stroke. In this particular line 59·7 per cent. of the

temperature fall due to cooling loss is incurred in the first 0·3 of the stroke, and only 40·3 per cent. in the last 0·7 of the stroke.

Comparing the two portions of the expansion line D E in the same manner, it is found that 59·3 per cent. of the total fall is incurred on the first 0·3 and 40·7 per cent. in the last 0·7 of the stroke.

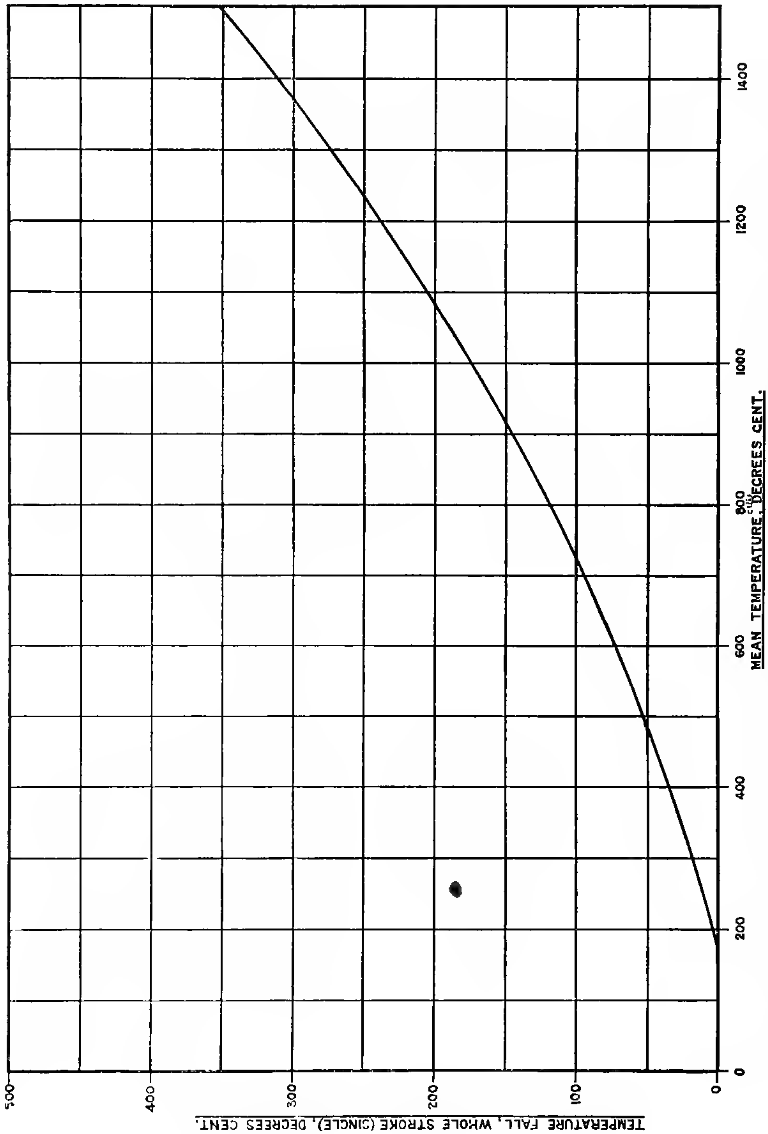


FIG. 86.—Temperature fall for complete single strokes plotted against mean temperature during the experiment for three cards under full-load conditions similar to fig. 81.

In these two expansion lines the temperature fall may be taken as 60 per cent. in the first 0·3 and 40 per cent. in the last 0·7.

Comparing the two parts of FG in this manner, it is found, however, that 48·5 per cent. temperature fall is incurred in the first 0·3 and 51·5 per cent. in the last 0·7.

Arranged in tabular form, the particulars for these three expansion lines are as follows :

		Mean temp. in time	Temp. fall due to heat loss
1	Complete expansion line BC	930° C.	155° C.
	Partial expansion line BB′	1190° C.	92°·5 C.
2	Complete expansion line, DE	690° C.	93° C.
	Partial expansion line DD′	900° C.	55° C.
3	Complete expansion line FG	560° C.	65° C.
	Partial expansion line FF′	700° C.	31°·2 C.

Lines 1 and 2, division of temperature fall 60 per cent. in first $\frac{3}{10}$; 40 per cent. in last $\frac{7}{10}$ of stroke;
and line 3, say, 50 per cent. in first $\frac{3}{10}$; 50 per cent. in last $\frac{7}{10}$ of stroke.

The division is practically the same in the first two lines, but the ratio alters in the third, where the mean temperatures fall enough to approximate more closely to the temperatures of the valves and piston surface. Where this occurs the mean temperature of the enclosing walls is so much higher at $\frac{3}{10}$-stroke as to affect the law of cooling. That this is so will be seen from experiments which give approximate determinations of the actual mean temperatures of the cylinder walls at different parts of the stroke.

These figures clearly show that in these expansion curves at the higher mean temperatures a larger temperature fall is incurred at the first 0·3 of the stroke as compared with the remaining 0·7, notwithstanding the fact that a greater surface is exposed in the latter part. The difference between the distribution in the first and third lines shows, however, that the distribution must not be taken for granted—it will vary with many circumstances. From these curves it is possible to reason upon the explosion expansion line in order to arrive at an approximate value of the temperature fall distribution throughout the stroke.

By dealing with the card shown at fig. 81, and measuring the mean temperature in time of the explosion and expansion line *bc*A, see fig. 84, then that of the explosion and partial expansion line *bc*A^2, using curves figs. 85 and 86, the following values will be obtained :

Mean temperature on explosion expansion line	=	1250° C.
Temperature fall corresponding	=	257° C.
Mean temperature on explosion and partial expansion line to $\frac{3}{10}$	=	1410° C.
Temperature fall corresponding . . $\frac{256}{2}$	=	128° C.

That is—

Temperature fall on first 0·3 of stroke = 128, say 50 per cent.
and „ „ last 0·7 „ 129, say 50 per cent.

In this case the temperature fall is equally divided: half is incurred in the first 0·3 and half in the last 0·7 of the stroke. If the ignition had been more rapid, however, so that the maximum pressure was attained before the piston had moved out appreciably, then the distribution would have been different, because the mean temperature of the explosion expansion line for the first $\frac{3}{10}$ of the stroke would have been higher. Every variation in the explosion curve will produce a corresponding variation in the distribution of temperature fall.

To compare the effect of the increased surface due to the outward position of the piston only, it is necessary to compare the temperature falls incurred in equal times and equal temperatures for the $\frac{3}{10}$ piston movement and the whole piston movement of a complete out stroke.

To do this a considerable number of diagrams have been measured for 120 revolutions per minute with the engine cold—that is, no load upon it, and only ignitions sufficient to keep it running at the speed while the water was freely passed through the jacket at a temperature of about 13° C.—and also for 160 revolutions with the jacket hot, water leaving about 80° C., while the engine carried a load of 50 BHP.

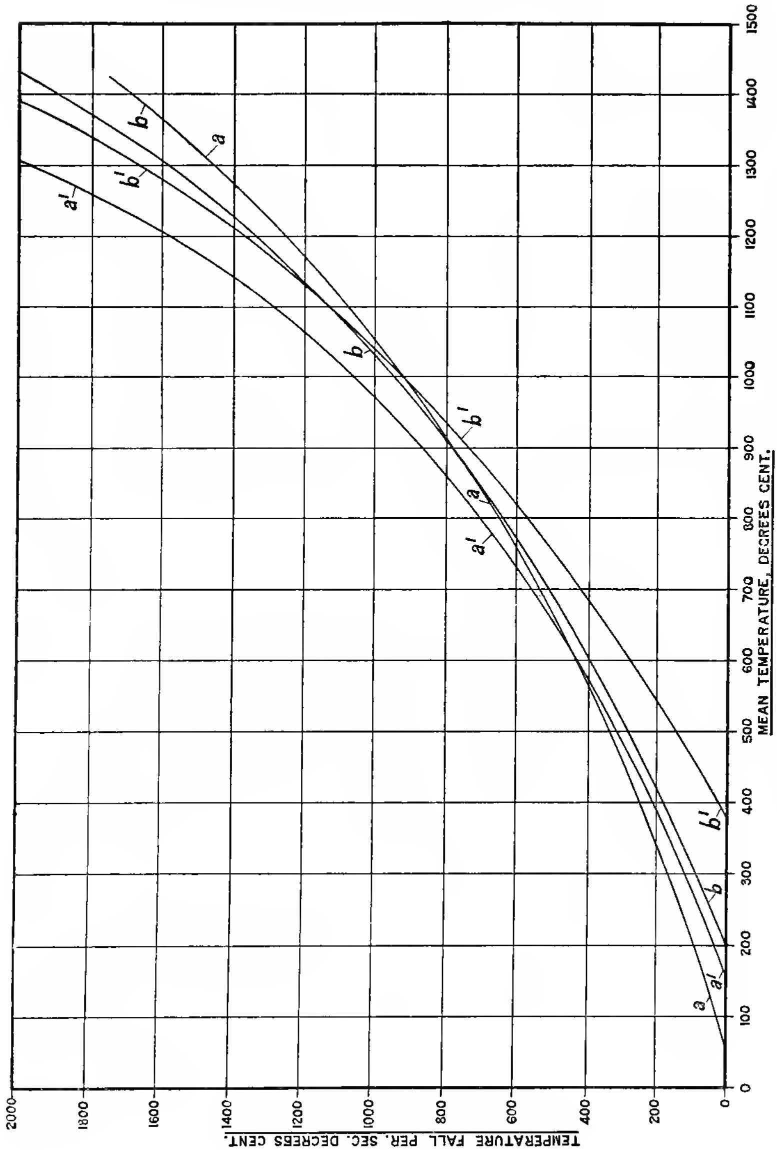
The curves shown at fig. 87 give the temperature falls incurred per second for different mean temperatures calculated in time. The particulars are fully given under the diagrams.

Consider, first, the curves *a a'*; these correspond to the conditions in the engine cylinder when the engine is running at 120 revolutions per minute without any load, so that very few ignitions keep it in motion, and while the water jacket is kept cold by running water at 13° C. freely through it. Under these conditions the end of the piston and the interior surface of the valves will be but little heated by the explosions, and the interior surfaces of the cylinder and combustion space wall will tend to fall between the explosions to the water-jacket temperature of 13° C.

The curve *a* is that for the complete strokes, while *a'* is for the partial first $\frac{3}{10}$ of the stroke. Taking *a* first, it appears that the rate of temperature fall per second for a mean temperature of 1300° is 1460° C.; that is, that a complete stroke of the engine during which the mean temperature in the cylinder was 1300° C. would lose heat to the walls at a rate sufficient to produce a temperature fall of 1460° C. per second. Of course the stroke does not last a second, but only

a quarter of a second, so that the real temperature fall under these conditions would be $\frac{1460}{4} = 365°$ C. It is better, however, to express the temperature fall in degrees per second.

On this curve the following mean temperatures—1200°, 1100°,



120 revs. per min. Cold {(a) Whole stroke. (a') Upper $\frac{1}{10}$ stroke.

160 revs. per min. Hot {(b) Whole stroke. Load 50 B.H.P. {(b') Upper $\frac{1}{10}$ stroke.

FIG. 87.—Temperature fall incurred per second at different mean temperatures calculated in time. Each line is mean of three cards under the given conditions

1000°, 900°—give respectively 1250°, 1080°, 920°, and 780° C. temperature fall per second.

It will be observed that the curve is concave, so that the rate of temperature fall increases with the increase of temperature.

This curve, when prolonged to the zero line of temperature fall, cuts it at the temperature of 65° C., which means that when the gases within the cylinder fall to the mean temperature of 65° C., no further heat loss occurs to the enclosing walls—that is, the mean temperature of the enclosing walls for the complete stroke of the engine must be 65° C. The water outside does not succeed in keeping down the mean temperature of the interior surface, including valve surfaces and piston end, to its own temperature of 13° C. This curve thus gives an interesting indication of the temperature of the walls.

Taking curve *a'* in the same way, its general slope is greater than *a*, and it cuts the zero line at 165° C., so that, although its rate of change is greater, it does not cross the curve *a* until a mean temperature of 610° C. is attained. Up to this point the temperature fall is less for a given mean temperature, above it the temperature fall rapidly increases with increasing mean temperature, so that the rate of fall is considerably greater in *a'* than in *a* at the higher mean temperatures of 1200° and 1300° C.

The fact that the intersection with the zero line is at 165° indicates that the mean temperature of the cylinder walls is much higher for the inner $\frac{3}{10}$ of the stroke than for the whole stroke. This was to be expected, because the proportion of piston end, valve surface, and other unjacketed surfaces becomes greater as the piston moves in—that is, the water-jacketed cylinder surface is covered as the piston moves in, so that the ratio changes, and therefore the mean temperature rises. Apart from this, however, it was to be expected that the surface temperature of the jacketed and unjacketed parts would be highest at the combustion chamber end, where it is exposed to the maximum temperature. This curve, then, indicates that the mean temperature of the $\frac{3}{10}$ surface during the $\frac{3}{10}$ period is 165° C., while the mean temperature of the whole cylinder surface during the whole stroke period is only 65°, and this with the water in the water-jacket at 13° C.

Compare now the two curves *a* *a'* at the mean temperatures following:

Mean temperature	1300° C.	1200° C.	1100° C.	1000° C.	900° C.
Temperature fall per sec., curve *a*	1460	1250	1080	920	780
Temperature fall in 0·25 sec., curve *a*	365	312	270	230	195
Temperature fall per sec., curve *a'*	1970	1570	1285	1060	870

The rate of loss of temperature is less in curve *a* by the following percentages, calculated on the *a* values for the successive temperatures :

nearly 26 per cent., 20·5 per cent., 16 per cent., 13 per cent., and 10 per cent.

This clearly shows that, notwithstanding the diminutions of surface exposed in a given time at $\frac{3}{10}$ stroke, as compared with whole stroke, the absolute temperature fall rate is increased. This is probably due to the fact that the mean surface exposed diminishes more slowly than the mean density increases, so that the economical effect of increased density, discussed in a previous chapter, is not realised. The existence of some turbulent motion may also mask the other effects. This clearly shows that it is necessary to determine cooling in the cylinder with the moving piston, as the conditions are too little known to be entirely predicted from explosions in closed vessels of fixed capacity. This becomes even more evident when the curves *b b′* are studied. These curves *b b′* are taken while the engine is running at a load of 50 BHP. Here the explosions were almost consecutive and the water-jacket temperature was 80° C., so that the interior surfaces, both jacketed and unjacketed, were much hotter than in curve *a a′*; as before, curve *b* is for whole stroke and *b′* for first $\frac{3}{10}$ of stroke. Curve *b* cuts the zero line at 190°, indicating the mean temperature of the enclosing walls during the whole stroke. The condition of the internal surfaces is different—190°, as compared with 65°, a very large increase in temperature. Curve *b′* cuts the zero line at 400°, also a much higher temperature. The continued explosions and the hot-water jacket have produced a very considerable change upon the surface temperature of the enclosing walls. Here also the curve *b* for the whole stroke is less steep than *b′* for the $\frac{3}{10}$ stroke, but, owing to the high wall mean temperature at $\frac{3}{10}$, they do not cross till the temperature of 1100° C. is passed. Below this temperature the temperature fall is less in *b′*; above that temperature it is greater.

Comparing *b b′* as *a a′* have already been treated at the same mean temperatures :

Mean temperature	1300° C.	1200° C.	1100° C.	1000° C.	900° C.
Temperature fall per sec., curve *b*. .	1580	1340	1130	950	780
Temperature fall per 0·187 sec., curve *b*	290	250	211	177	146
Temperature fall per sec., curve *b′* .	1650	1370	1125	920	730

For 1300° and 1200° C. the temperature fall is greater in *b′* than in *b*, but only 4 per cent. and 2 per cent. respectively, calculated on *b′*; at 1100° C. the temperature fall is practically equal; at 1000° *b′* is less by about 3 per cent. on *b′*, and 900° C. is nearly 7 per cent. less, also calculated on the value of *b′*.

It is thus seen that for the practically interesting range of mean temperatures 1300° to 900° C. the temperature fall given by the two curves varies but little. That is because of the high average

temperature of the interior surface. The point of intersection of the curves *b* and *b′* is raised to such a high value that the actual heat loss about these temperatures remains nearly constant, although the two curves are undoubtedly different in their slope.

The temperature fall on *b* is also given for 0·187 second, as this is the period of one stroke of the engine at 160 revolutions per minute. These numbers are therefore the temperature falls incurred during one forward stroke due to heat flow to the walls under working conditions.

Compare now the curves *a′ b′*, and it will be seen that they are fairly parallel one to the other. Their general slope is very similar, so that, except at the lower end, they could almost be superimposed. It appears as if the difference in absolute value of the temperature fall for given mean temperatures were due mainly to the difference between the enclosing wall temperatures.

These experiments prove conclusively that cooling behind a moving piston depends largely on the varying temperature of the water jacket, and still more upon the varying mean temperatures of the cylinder walls according to the condition of load and water circulation. And although general laws may be deduced from closed vessel experiments of fixed volume at varying initial pressures, yet the problem in the working engine is so complex that it is desirable to make many direct determinations on actual engines as to explosion and cooling in cylinders of varying dimensions before endeavouring to deduce any general formulæ.

It is useful, however, to compare the temperature falls so observed with those in closed vessels of initial atmospheric pressure given in the previous chapter at the working condition of heavy load, and for that take the curve *b* full stroke at the following mean temperatures, reducing the temperature falls to $\frac{1}{20}$ second instead of 1 second, and calculating the cubic foot degrees per square foot at the temperature of charging as given below :

Mean temperature	1400° C.	1300° C.	1150° C.
Temperature fall in $\frac{1}{20}$ sec.	93·5	79	61·5
Cubic foot degrees per sq ft. surface, piston assumed full out	20	17	13

The capacity of the cylinder and combustion space when the cylinder is full out is 2·41 cub. ft., and the surface exposed is 11·2 sq. ft., so that cubic foot degrees are obtained by multiplying the respective temperature falls by $\frac{2·41}{11·2} = 0·215$. The temperature fall values obtained from the cooling curves of Hopkinson's large vessel of 6·2 cub. ft. capacity (see p. 204) in cubic foot degrees for the same three mean temperatures are: 24·4 – 18·6 and 13·6.

so that the temperature falls here given closely resemble those which would have been obtained in a closed vessel of 2·41 cub. ft. capacity exposed for the same time to the cooling walls.

It is to be noted that the absolute value of the cubic foot degree used in the moving piston experiments is less than the cubic foot degree in the closed vessel experiments, because the mean temperature of the engine charge is 95° instead of 16° C., which makes the value for the former $\frac{273 + 16}{273 + 95} = 0{\cdot}79$ that of the latter.

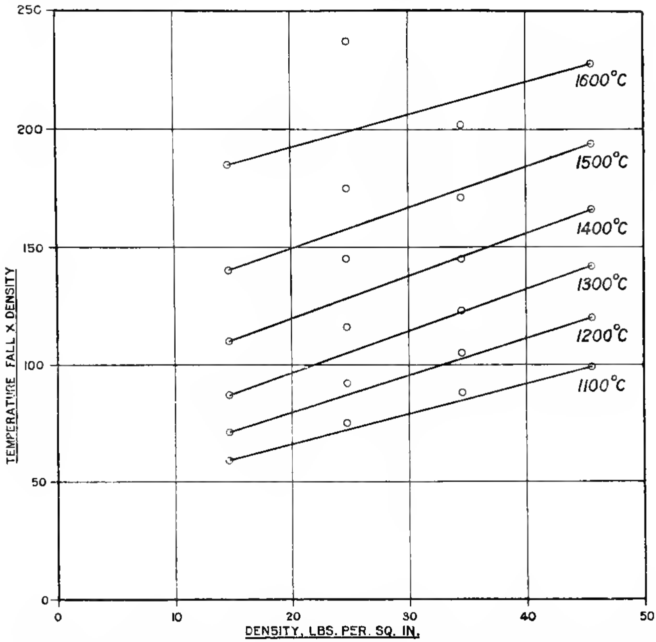


FIG. 88.—Proportional heat flow at different densities and temperatures, calculated from Bairstow and Alexander's Experiments

This does not, however, affect the reasoning as to relative cooling, but applies only where absolute rates of heat flow are to be compared. It has been shown that absolute heat flow to the enclosing walls increases with the density of the gas exposed to unit surface for unit time, but for any given space the flow increases at a slower rate than the density, so that at higher densities the temperature fall rate is less than at low densities. Broadly, the absolute heat flow increases from 1 to 1·5 when the density increases from 1 to 3 at temperatures of 1200° and 1300° C., such as are common mean temperatures for the working stroke of a gas engine.

To understand the effect of the moving piston on the cooling curve, it is desirable to formulate approximately the effect of density upon absolute heat flow through the walls. This can only be attempted on a rough approximation in the present state of experimental knowledge, but it is useful to attempt it.

The curves shown at fig. 88 have been prepared from the table at p. 208, as deduced by the author from Bairstow and Alexander's experiment. The temperature falls there shown are those occurring in the explosion vessel at varying pressures before ignition, so that the actual heat flow for a given temperature fall is greater at the higher initial pressures. To compare the absolute heat flow at these temperatures and pressures, it is necessary to convert the numbers by taking density into account for the pressures 24·7, 34·5, and 44·8 lbs. absolute; the temperature falls corresponding have been multiplied by $\frac{24·7}{14·55}$, $\frac{34·5}{14·55}$ and $\frac{44·8}{14·55}$. Taking the lower pressure, 14·55 lbs. per sq. in., as the atmospheric pressure at the time of the observations, the numbers so obtained give the absolute, not the relative, heat losses.

They are plotted against density in pounds per square inch. It will be observed that the results through the observations for 1100°, 1200°, 1300°, 1400° and 1500° C. are represented by straight lines, and that these lines are all nearly parallel to each other. The observations require repetition to eliminate some discrepancies, but broadly they are properly represented within the limits of accuracy of the experiments.

These values of temperature fall multiplied by density are represented approximately by the formula

$$t\,d = t + (d - 1)\,25,$$

Where t = temperature fall at the mean temperature during $\frac{1}{20}$ second ignited at atmospheric pressure;

d = density, in atmospheres and 25 the value of a constant;

$t\,d$ = the required value at the density d in atmospheres.

The heat flow in cubic foot degrees per square foot at atmospheric pressure and temperature of the experiments is found from this by the formula

$$\text{H}f = \frac{(t + (d - 1)\,25)\,c}{s} = \frac{tc}{s} + \frac{(d - 1)\,25c}{s}$$

where Hf = heat flow per square foot exposed to walls in cubic foot degrees at atmospheric pressure and temperature;

t = temperature fall in vessel in $\frac{1}{20}$ second at atmospheric density and temperature;

d = density before ignition in atmospheres;

c = capacity of vessel in cubic feet;

s = surface exposed in square feet.

Bairstow and Alexander's vessel was of 0·82 cub. ft. capacity and 5·02 sq. ft. internal surface exposed. Assume that it is desired to compare heat flow at 3 atmospheres initial pressure with 1 atmosphere. It will be observed that the expression $\frac{(d-1)\,25\,c}{s}$ does not include t, so that it is independent of that temperature fall, where $d = 3$, $c = 0·82$, and $s = 5·02$

$$\frac{(d-1)\,25\,c}{s} = 8·1$$

t for 1000° C. is 49, so that

$$\frac{tc}{s} = \frac{49 \times 0·82}{5·02} = 8.$$

The loss per square foot at 1000° at atmospheric pressure is 8, see table p. 212, and at 3 atmospheres it is 16·1. At 1300° C. atmospheric density loss 14·2; at 3 atmospheres $14·2 + 8·1 = 22·3$. Assuming the formula to hold at 6 atmospheres, the value for 1300° C. is $14·2 + 20·2 = 34·4$ cubic foot degrees.

It is now possible to consider the effect of the moving piston, and it is desirable to take as a case the cylinder of the National Gas Engine Company's engine which has been discussed.

This cylinder is 22 ins. stroke and 14 ins. diameter. When the piston is full out, the total interior surface is 11·2 sq. ft.; when the piston is full in, the surface enclosing the explosion space, including the surface of the piston end, is 4·5 sq. ft. The cylindrical surface swept by the piston is 6·7 sq. ft. Assume the compression ratio to be $\frac{1}{6}$ and the density of the charge when fully compressed 6 atmospheres. If the piston remain full out, and a charge of gas and air be fired at atmospheric pressure, the piston remaining fixed while cooling goes on; our experiments prove that the heat flow for 1300° C. mean temperature during $\frac{1}{20}$ second will be 14·2 cub. ft. degrees per square foot. That is, the cylinder full of gases will lose $14·2 \times 11·2 = 159$ cub. ft. degrees in $\frac{1}{20}$ second. If the piston remain full in, and a charge of gas and air be fired at a density of 6 atmospheres, the piston remaining fixed while cooling goes on, our numbers show that the heat flow for 1300° C. mean temperature during $\frac{1}{20}$ second will be 34·4 cub. ft. degrees per square foot. That is, the cylinder full of gases will lose $34·4 \times 4·5 = 154·8$ cub. ft. degrees, but little less than that lost to the larger surface in the same time. If the increase or diminution of surface exposed in an engine cylinder followed the inverse law of the diminution or increased flow due to density, then, so far as surface and density change affected the matter, the same absolute heat loss would be suffered by a charge from equal mean temperature at every point of the stroke. Surface always increases with diminution of density, so that the changes cancel

out to some extent, but the laws of surface change vary with the proportions of the cylinder diameter and stroke, and also with the configuration of the explosion space, so that engines must differ in this respect. Diagrams could be constructed, however, for any engine on a time base dividing up to the period of the stroke into ten parts, and mean surface, mean density, and mean temperature could be calculated for each part, so that heat flow could be calculated for the whole stroke in any particular case. The data available from closed vessel experiments are not yet sufficiently complete and concordant to enable such calculations to be carried further with advantage to the engineer.

The Clerk diagram, when more fully studied and applied, will enable full information to be ultimately obtained.

So far, the measure of heat quantity used in the present and two preceding chapters is the cubic foot degree, and no attempt has been made to deduce its value in any standard heat unit or in foot-pounds. Not only is its value unknown, but the question as to variation with temperature is so far left open.

If a gas be compressed without gain or loss of heat from volume V_o to V_1, and the temperature rises from T_o to T_1, so that the work done upon the gas is W, then the mean specific heat C_v of the gas per unit volume at 0° and 760° mm. at constant volume between the temperatures is:

$$C_v = \frac{W}{\psi_o(T_1 - T_o)}$$

where ψ_o is a constant depending on the quantity of the gas in the cylinder.

This is also true of expansion as well as compression. The dynamical value of the rise or fall of 1° C. for 1 cub. ft. of the gas will be given by the same formula:

$$D_v = \frac{W}{V(T_1 - T_o)}$$

where W is the work done by or on the gas in foot-pounds, V is the volume in cubic feet, and D_v is the dynamical value in foot-pounds.

It is evident that the method of operation described in this chapter affords the means of determining the heat loss on expansion or compression lines, and so permits the temperature fall or rise, due to work done, to be determined at any temperatures. In this way the author has experimented on the products of combustion contained within a gas-engine cylinder, and has deduced the values of D_v for that working fluid at different temperatures.

The experiments were numerous, and many difficulties were encountered, for the full discussion of which the reader is referred to the

paper describing the experiments read by the author before the Royal Society in 1906.[1]

It is sufficient here to give the values obtained for the working fluid, which was of the following composition :

Steam (assumed gaseous) . .	11·9 volumes
Carbon dioxide	5·2 ,,
Oxygen	7·9 ,,
Nitrogen	75·0 ,,
	100·0 volumes.

TABLE OF APPARENT SPECIFIC HEATS (INSTANTANEOUS) IN FOOT-POUNDS PER CUBIC FOOT OF WORKING FLUID AT 0° C. AND 760 MM.

Temperature	Specific heat at constant volume	Temperature	Specific heat at constant volume
° C.	ft.-lbs.	° C.	ft.-lbs.
0	19·6	800	26·2
100	20·9	900	26·6
200	22·0	1000	26·8
300	23·0	1100	27·0
400	23·9	1200	27·2
500	24·8	1300	27·3
600	25·2	1400	27·35
700	25·7	1500	27·45

TABLE OF MEAN APPARENT SPECIFIC HEATS IN FOOT-POUNDS PER CUBIC FOOT OF WORKING FLUID AT 0° C. AND 760 MM.

Temperature	Specific heat at constant volume	Temperature	Specific heat at constant volume
° C.	ft.-lbs.	° C.	ft.-lbs.
0—100	20·3	0—900	23·9
0—200	20·9	0—1000	24·1
0—300	21·4	0—1100	24·4
0—400	21·9	0—1200	24·6
0—500	22·4	0—1300	24·8
0—600	22·8	0—1400	25·0
0—700	23·2	0—1500	25·2
0—800	23·6		

These tables enable a definite value to be given to the cubic foot degree of gas-engine mixture, which has been dealt with in this chapter, and they show that for various reasons the apparent specific heat values increase considerably with temperature. These numbers will be used later in this work, together with the other data arrived at in this chapter.

[1] *Proceedings of the Royal Society*, A, vol. 77, 1906, p. 499.

CHAPTER IX

THERMAL AND MECHANICAL EFFICIENCY OF THE DIFFERENT TYPES OF GAS ENGINE IN USE

In the previous chapters the different types of gas engine have been considered as air engines—that is, engines in which the working fluid is atmospheric air of constant specific heat throughout the temperature range, and the theoretic efficiencies under varying conditions have been calculated and compared on this assumption. The phenomena of gaseous explosion have also been studied, and the properties of the working fluid, the result of a gaseous explosion, have been discussed. The way is now clear for the study of the results obtained from the engines in practice, and in this chapter it is proposed to discuss, so far as is possible, tests made by observers independently of the engine constructors. It is proposed to deal with standard tests mainly.

Type I

The most important engines of this type which have been in public use are those of Lenoir and Hugon and their mechanical arrangements have been sufficiently described in the Historical Sketch. As they have long disappeared from practice, it is undesirable to deal with their results at any length. It is sufficient to say that Professor Tresca's experiments made in Paris with a Lenoir engine nearly fifty years ago showed an indicated efficiency of only 4 per cent. That is, the engine received, say, 100 heat units as inflammable gas, and the indicator diagrams obtained showed that only 4 heat units of the hundred appeared as work done by the expanding gases upon the piston. For the conditions of the diagrams obtained by Tresca, and later diagrams obtained by Slade, it is found that a corresponding air-engine diagram without heat losses would give 17·5 per cent. down to 12·6 per cent. efficiency, so that the losses in the Lenoir engine were undoubtedly very great. The theoretical air-engine efficiency was poor, and its realisation in practice was still poorer.

Later experiments by Professor Tresca on the Hugon engine showed somewhat better results.

Experiments were made by the author in 1884 upon a Bischoff engine, and the indicated thermal efficiency was found to be somewhat lower than 4 per cent., Professor Tresca's number.

Type IA

The most important engines of this type were those of Barsanti and Matteucci, 1857, and Otto and Langen, 1866; they are found fully described in the Historical Sketch. The Otto and Langen engine attained considerable success in practice, and the author made a number of tests with it in 1885. The engine tested was of 2 HP; its cylinder was 12·5 ins. diameter, and its longest observed stroke was 40·5 ins.; working at the rate of 28 ignitions per minute, the indicated power was 2·9 horse; the consumption of Oldham gas was 24·6 cub. ft. per IHP hour. The BHP was 2 horse, and the brake consumption was 36 cub. ft. per BHP hour. This did not include the consumption of the gas-supply to the igniting flame, which was 12 cub. ft. per hour.

The thermal efficiencies were as follows:

Indicated thermal efficiency	=	16 per cent.
Brake thermal efficiency	=	11 ,,

The largest engine built of this type was of 3 HP, and it is probable that its brake consumption was about 30 cub. ft. of Oldham gas per BHP hour, which would be equivalent to:

Brake thermal efficiency .	13 per cent.

Professor Tresca tested a half-horse engine at the Paris Exhibition in 1867; it gave 0·456 brake horse and consumed Paris gas at the rate of 44 cub. ft. per BHP hour. The gas consumed by the igniting flame is not included in this figure.

It was a great matter to realise so much as 16 per cent. of the total heat supplied in indicated work and 11 per cent. in brake work at the early date of 1867; but the cumbrous nature of the engine and its noisy action prevented its general adoption, although its historic interest is great. The theory of the action of this engine is somewhat fully discussed in the earlier editions of this book, but its present interest does not warrant further mention here.

Type II

In engines of this kind compression is used previous to ignition, but the ignition is so arranged that the pressure in the motor cylinder does not become greater than that in the compressing pump. The power is generated by increasing volume at constant pressure. Engines of Type II are therefore:

Engines using a mixture of inflammable gas and air compressed before ignition and ignited in such a manner that the pressure does not increase, the power being generated by increasing volume.

These engines are truly slow-combustion engines; in them there is no explosion.

The two engines which found some practical use between 1876 and 1880 were those of Brayton and Simon. They are sufficiently described in the Historical Sketch. The only engine of Type II now in public use is that of Diesel, and its use so far has been confined to oil engines. As it uses the Otto or four-cycle system of mechanical operation, it is proposed to discuss its thermal and mechanical efficiency after the Otto cycle engines in the consideration of Type III. Early tests were made, however, by the author on a Brayton engine, which will now be described.

The author's test was made at the Crown Ironworks, Glasgow, on February 21 and 22, 1878.

The engine was made by the New York and New Jersey Ready Motor Company. The motor cylinder was 8 ins. diameter and the stroke 12 ins.; the pump cylinder was also 8 ins. diameter, and the stroke 6 ins. The details of the engine tested are shown at figs. 9, 10, 11, and 12 in the Historical Sketch.

Diagrams were taken from both pumps and motor by a well-made Richards indicator. At the same time the brake was applied to the fly-wheel, fully loading the engine; readings were taken at regular intervals. The revolutions were recorded by a counter. The petroleum used was measured in a graduated glass vessel.

The results are as follows:

Test of Brayton Petroleum Engine 1878. (*Clerk*)

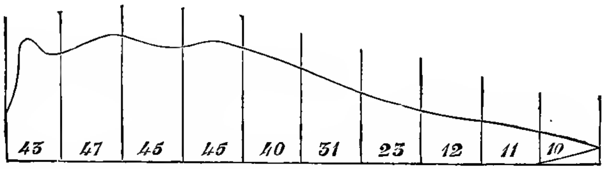
Petroleum consumed during one hour	1·378 gallon
Mean speed of engine	201 revs. per minute
Mean brake reading	4·26 HP
Mean pressure, power cylinder	31 lbs. per sq. in.
Mean pressure, air pump	27·6 lbs. per sq. in.
Piston speed, motor	201 ft. per minute
Piston speed, pump	100·5 ft. per minute
Power indicated in motor	9·49 HP
Power indicated in pump	4·10 HP
Available indicated power	5·39 HP

The power by the dynamometer is 4·26 horse; therefore the mechanical friction of the engine is $5{\cdot}39 - 4{\cdot}26 = 1{\cdot}13$ horse.

Consumption of petroleum	0·255 galls. per IHP per hour
Consumption of petroleum	0·323 galls. per actual HP per hour

Figs. 89 and 90 are diagrams from the motor and pump, which are fair samples of those taken. It will be observed that considerable throttling occurs in entering the motor cylinder; the pump pressure is higher than the reservoir pressure, and the motor pressure is lower,

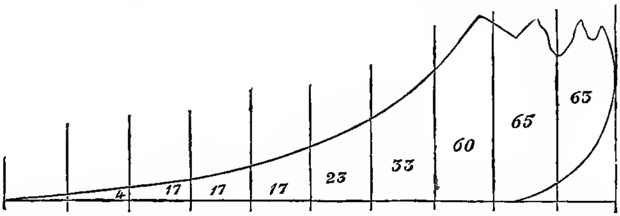
so that a double loss has been incurred. The principle of the engine is so good that the author anticipated better results. Great improvements could be obtained by reproportioning the valves and air-passages; they are in this engine much too small and cause needless resistance and loss. The maximum pressure in the motor cylinder is 48 lbs. per square inch, which remains steadily till the inlet valve shuts at four-tenths of the stroke: the pressure then slowly falls as the gases expand, the exhaust valve opening at about 10 lbs. per square inch above atmosphere.



Mean pressure 30·2 lbs. per sq. in. 8 ins. dia. cylinder. Stroke 12 ins. 200 revs. per min.

FIG. 89.—Brayton Petroleum Engine. Motor Cylinder

The average available pressure upon this diagram is 30·2 lbs. per square inch. The air pump shows a maximum pressure of 65 lbs. per square inch, the reservoir pressure being 60 lbs. The average resistance is 27·6 lbs. per square inch; as the pump is half the stroke of the motor and equal to it in area, the pressure to be deducted is $\frac{27 \cdot 6}{2} = 13 \cdot 8$ and $30 \cdot 2 - 13 \cdot 8 = 16 \cdot 4$. The actual available pressure actuating the engine is therefore only 16·4 lbs. per square inch. The



Mean pressure 27·6 lbs. per sq. in. 8 ins. dia. cylinder. Stroke 6 ins. 200 revs. per min.

FIG. 90.—Brayton Petroleum Engine. Pump Cylinder

effect of the clearance in the pump cylinder is noticeable upon the diagram; the air-inlet valve does not open till one-tenth of the down-stroke is completed.

The theoretical efficiency of this type, with a maximum temperature of 1600° C., compression of 60 lbs. per square inch above atmosphere, and motor cylinder of twice the pump volume, is 0·30.

But the diagram is imperfect in many ways. Using the mixture it does, the diagram should show a maximum temperature of 1600° C. at least; in reality, the highest temperature is only 840° C. The flame is entering the cylinder at an actual temperature of 1600° C. during the whole period of admission, but the convection has so greatly increased by the mixing effect of the entering current that greater cooling results; accordingly, when the gases are fully admitted and the inlet valve is closed, the gases have only a temperature of 840° C. instead of 1600° C. After admission ceases, the expansion line from 45 lbs. to 10 lbs. pressure is far above the adiabatic, indeed it is isothermal, the combustion is proceeding and the small igniting flame also is helping to sustain the temperature.

It is therefore quite evident that the loss of heat is much greater than that occurring during explosion in equal time.

The specific gravity of the petroleum was 0·85, therefore the weight of one gallon is 8·5 lbs. As 0·255 gallon is burned per IHP per hour, this amounts to $8·5 \times 0·255 = 2·16$ lbs. of liquid fuel per IHP per hour. One pound gives out 11,000 heat units, and for one horse-power for one hour 1,424 units are required; the actual indicated efficiency is therefore

$\frac{1{,}424}{2·16 \times 11{,}000} = \frac{1{,}424}{23{,}760} = 0·06$ nearly; that is, 6 per cent. of the whole heat given to the engine is accounted for by the power developed in the motor cylinder.

If there were no losses of heat to the cylinder, or losses by throttling during the inlet and transfer of the air from the pump to the motor, or loss of heat from the reservoir to the atmosphere, then the efficiency of this type of engine would be 30 per cent. These losses in practice reduce it to 6 per cent. The cycle is a good one, and under other circumstances is capable of better things, but it is unsuitable in this form for a cold-cylinder engine. Cooling and undue resistance are the main causes of the great deficit.

The gases entering the cylinder as flame in passing through the inlet chamber expose a large surface to the action of the water jacket; the entering currents also impinge against the piston, causing more rapid circulation than ordinary convection. Both causes intensify the cooling action of the cylinder walls. In the engine tested by the author the communicating pipes and the motor admission valve were much too small; a considerable loss of pressure resulted; although the reservoir pressure was 60 lbs., that in the cylinder never exceeded 48 lbs. above atmosphere, showing a loss of 12 lbs. per square inch from undue resistance. To enable this engine to realise the advantages of its theory considerable modifications in its arrangements are required.

Type III

Engines of this kind resemble those just discussed in the use of compression previous to ignition, but differ from them in igniting at constant volume instead of constant pressure; that is, the whole volume of mixture used for one stroke is ignited in a mass instead of in successive portions.

The whole body of mixture to be used is introduced before any portion of it is ignited; in the previous type the mixture is ignited as it enters the cylinder, no mixture being allowed to enter except as flame. In Type III the ignition occurs while the volume is constant; the pressure therefore rises; it is an explosion engine in fact, like the first type, but with a more intense explosion due to the use of mixture at a pressure exceeding atmosphere.

The most obvious means of applying the method is that suggested by the Lenoir engine. The addition of a pump taking mixture at atmospheric pressure, compressing it into a reservoir from which it passes to the motor cylinder at the increased pressure, seems a simple matter. The igniting arrangements would act as in the original. As the gases are under pressure, the piston would take its charge into the cylinder in a smaller proportion of the forward stroke, and so more of the motor stroke would be available for useful effect. The diagram such an engine should produce is seen at fig. 26, p. 74; the shaded part is the available portion, the other part is the pump diagram. The theoretic efficiency of such an engine is as good as the type can give. The patent list shows that it was the first proposed after Lenoir. Many such engines have been attempted and have given very good results economically, but the difficulties of detail are considerable, the greatest being the necessity for the intermediate reservoir. Million's patent 1861 proposes to do this, the present author also constructed one of this kind in 1878, and later one was made by Mr. Atkinson. The difficulties, however, are too great to allow the success of small motors on the plan.

Mr. Otto, the first to succeed with the free piston engine, was also the first to succeed in adapting compression in a reliable form.

In the third type are included all engines having the following characteristics, however widely the mechanical cycle may vary: Engines using a gaseous explosive mixture, compressed before ignition, and ignited in a body, so that the pressure increases while the volume remains constant. The power is obtained by expansion after the increase of pressure.

In the Otto or four-cycle gas engine, the first to combine the

compression principle with a simple and thoroughly efficient working cycle, the difficulties of compression are overcome in a strikingly original manner. To the engineer accustomed to the steam engine, the main idea seems a bold and indeed a retrograde step. The early gas engines were moulded more upon the steam engine model and were to some extent double acting. The Lenoir and Hugon both received two impulses for every revolution, the Brayton was single acting, and the Otto is only half single acting. The steam engine in its advance passed from single to double acting, and then to four and even more impulses per revolution. The gas engine in its progress has in this respect moved backwards, beginning with double action and then going back. The gain by this arrangement, however, has completely justified the retrogression.

A single cylinder serves alternately the purposes of motor and pump; during the first forward stroke of the piston, the valves are in such positions that gas and air stream into the cylinder from the beginning to the end of the stroke, the charge mixing as it enters with whatever gases the space may contain; the return stroke then compresses the uniform mixture into the space, and when the piston is full in, the pressure has increased to an amount determined by the relative capacity of the space. Meantime the charging valves have closed to the admission of gas and air, to permit of the compression, then ignition is caused when the compression is completed, the compressed charge ignites, and the pressure rises so rapidly that maximum is attained before the piston has moved appreciably on its forward stroke (second stroke); the piston is thus under the highest pressure at the beginning of its stroke and the whole stroke is available for the expansion.

This is the motive stroke. At the end of it, the exhaust valve opens and the return stroke is occupied in driving out the burned gases, except that portion remaining in the space which cannot be entered by the piston. These operations form a complete cycle, and the piston is again in the position to take in the charge required for the next impulse.

The cycle requires two complete revolutions, or four single strokes:

First out stroke.	Charging cylinder with gas and air.
,, in ,,	Compressing the charge into the space.
Second out stroke.	Explosion impelling piston.
,, in ,,	Discharging burned gases into atmosphere.

Thermal Efficiency of the Four-cycle Engines.—The Otto type engines are termed *four cycle* because the necessary operations require four piston strokes to complete the power-producing process.

The indicated thermal efficiency is determined by the proportion

of the total heat given to the engine, which appears as work done by the expanding gases upon the piston.

The brake thermal efficiency is determined by the proportion of the total heat given to the engine, which appears as work given out by the engine for overcoming external resistances.

In the early engines, of 100 heat units given to the engine in the form of coal gas 16 heat units appeared within the cylinder as work done by the expanding gases on the piston, so that the *indicated thermal* efficiency was 16 per cent. Such efficiencies were given by the Otto engines from 1877 to 1882. At the present day (1909) thermal efficiencies as high as 35 per cent. to 37 per cent. are readily obtainable. The indicated thermal efficiency has risen from 16 per cent. to 37 per cent. in thirty years; even higher efficiencies have been obtained experimentally, but 37 per cent. may be taken as above ordinary practice at the present time.

The following table of the results obtained by many experimenters will make this advance quite clear:

INDICATED AND BRAKE THERMAL EFFICIENCY OF FOUR-CYCLE ENGINES FROM 1882 TO 1908

No.	Mechanical efficiency	Names of experimenters	Year	Dimensions of engine	Indicated thermal efficiency	Brake thermal efficiency	Type of engine
	Per cent.			Diameter Stroke	Per cent.	Per cent.	
1	87·6	Slaby . . .	1882	6″·75 × 13″·7	16	14	Deutz
2	84·2	Thurston . .	1884	8″·5 × 14″	17	14·3	Crossley
3	86·1	Society of Arts .	1888	9″·5 × 18″	22	18·9	Crossley
4	80·9	Society of Arts .	1888	9″·02 × 14″	21	17	Griffin (6-cycle)
5	87·3	Kennedy . .	1888	7″·5 × 15″	21	18·3	Beck (6 cycle)
6	82	Capper .	1892	8″·5 × 18″	22·8	17·4	Crossley
7	87	Robinson .	1898	10″ × 18″	28·7	25	National
8	83	Humphrey .	1900	26″ × 36″	31	25·7	Crossley
9	81·7	Witz . . .	1900	51″·2 × 55″·13	28	22·9	Cockerill
10	85·5	Inst. Civil. Eng.	1905	14″ × 22″	35*	29·9	National
11	77·1	Burstall . .	1907	16″ × 24″	41·5†	32	Premier
12	87·5	Hopkinson .	1908	11″·5 × 21″	36·8	32·2	Crossley

* The value 35 per cent. is deduced by the author from the Inst.C.E. Committee's values.
† This value is, in the author's view, too high; probably due to indicator error.

It will also be observed that the brake thermal efficiency has increased from 14 per cent. to 32 per cent.; the 29·9—that is, 30 per cent. of the National Company's engine—represents their ordinary practice at moderate compression.

It has been shown in Chapter III that if the working fluid had been air, having a specific heat constant through the range, thermal efficiency would increase with increase of compression—that is, with diminution of the compression space in proportion to total volume behind the piston. Taking this ratio as

$$\frac{1}{r} = \frac{\text{compression space volume}}{\text{volume swept by piston + compression space volume}}$$

for the different values used in these engines and arranging the value of $\frac{1}{r}$ in the same order as the table with the indicated thermal efficiencies E_i below as follows:

No.	1	2	3	4	5	6	7	8	9	10	11	12
$\frac{1}{r}$	$\frac{1}{2·66}$	$\frac{1}{2·66}$	$\frac{1}{3·5}$	$\frac{1}{3·2}$	$\frac{1}{3}$	$\frac{1}{3·4}$	$\frac{1}{5·17}$	$\frac{1}{5}$	—	$\frac{1}{5·34}$	$\frac{1}{7·22}$	$\frac{1}{6·37}$
E_i	16%	17%	22%	21%	21%	22·8%	28·7%	31%	—	35%	41·5%	36·8%

Here it will be observed that, considered broadly, increased thermal efficiency is associated with diminution of the volume of the compression space, although it by no means follows that equal compression ratios give equal thermal efficiencies; in some cases even the thermal efficiency may be higher with the lower compression. These variations are to be expected from the great variations in the designs and dimensions of the twelve engines investigated. Broadly, however, other things being equal, increase in compression by diminution of compression space volume increases thermal efficiency.

If now these indicated thermal efficiencies be divided by the air standard efficiencies, as calculated in Chapter III corresponding to the various values of $\frac{1}{r}$, it will be seen that they have a well-defined relationship to that standard.

The values are again arranged as follows:

No.	1	2	3	4	5	6	7	8	9	10	11	12
$\frac{1}{r}$	$\frac{1}{2·66}$	$\frac{1}{2·66}$	$\frac{1}{3·5}$	$\frac{1}{3·2}$	$\frac{1}{3}$	$\frac{1}{3·4}$	$\frac{1}{5·17}$	$\frac{1}{5}$	—	$\frac{1}{5·34}$	$\frac{1}{7·22}$	$\frac{1}{6·37}$
Indicated thermal efficiency	16	17	22	21	21	22·6	28·7	31	—	35	41·5	36·8
Air standard efficiency	33	33	39	37	36	39	48	47	—	49	55	53
$\frac{\text{Indicated thermal efficiency}}{\text{Air standard efficiency}}$	0·48	0·51	0·56	0·57	0·58	0·58	0·60	0·66	—	0·71	0·75	0·69
Diameter of cylinder	6″·75	8″·5	9″·5	9″·02	7″·5	8″·5	10″	26″	—	14″	16″	11″·5

It will be seen that in the older engines the indicated thermal efficiency was 0·48 and 0·51 of that which an engine using air as the working fluid would have realised had heat losses and other imperfections of operation been entirely suppressed. That is, the Otto engines tested by Professor Slaby and Thurston gave half the indicated thermal efficiency which they would have done had they been perfect air engines performing exactly the same mechanical cycle of operations. This ratio gradually increases as the years pass, until the value becomes 0·75 in Professor Burstall's experiments. This value, however, the author considers to be too high. Professor Burstall's indicator appears to have erred in some way in this test, and the highest correct value appears to be 0·71, that deduced by the author from the Institution of Civil Engineers' experiments on a National gas engine of 14 ins. diameter cylinder and 22 ins. stroke.

From the foregoing it is obvious that indicated thermal efficiency increases with increasing compression: this fact clearly emerges from the examination of these twelve tests. It would be better, however, to determine this by experiments on the same engine in which the only variation made is a change in the volume of the compression space.

Professor Eugen Meyer, of Berlin, has made experiments of this nature with an engine of 7·8 ins. cylinder diameter and 11·8 ins. stroke, in which he arranged to vary the compression from 40 to 80 lbs. per sq. in. above atmosphere by means of a variation in the length of the connecting rod. The combustion space is thus changed in volume as required.

The following table has been calculated from Professor Meyer's experiments:

INDICATED THERMAL EFFICIENCY WITH VARYING COMPRESSIONS CALCULATED FROM PROFESSOR MEYER'S EXPERIMENTS

$\frac{1}{r}$	Indicated thermal efficiency	Air standard efficiency	Indicated efficiency / air standard	Revolutions per minute	Compression pressure above atmosphere	Dimension of engine, &c.
	per cent.	per cent.			lbs.	
$\frac{1}{4}$	25	44	0·58	257	80	Engine 7·8 ins. diam.
$\frac{1}{3{\cdot}8}$	24·4	42	0·58	249	75	by 11·8 ins. stroke.
$\frac{1}{3}$	21·4	37	0·58	251	50	Compression 40 to 80 lbs. per sq. in.
$\frac{1}{2{\cdot}7}$	18·8	33	0·57	225	40	above atmosphere

In Meyer's experiments it is evident that the indicated thermal efficiency bears a practically constant ratio to the air standard efficiency proper to the particular compression between 40 and 80 lbs. per sq. in. above atmosphere, or a variation of the compression ratio $\frac{1}{r}$ of from $\frac{1}{2{\cdot}7}$ to $\frac{1}{4}$ of the total volume behind the piston.

When the indicated thermal efficiency is 25 per cent., 24·4 per cent. and 21·4 per cent. of the total heat supplied to the engine, it is in the three cases 0·58 of that which would have been given if air alone had been the working fluid and it had suffered no change in its specific heat. When the indicated thermal efficiency is 18·8 per cent., it is 0·57 of the air standard. This small change may be due to the fact that the engine was running at 225 revolutions per minute during the test instead of about 250 revolutions, as in the other tests.

In these experiments the change from 40 lbs. to 80 lbs. compression raised the efficiency from 18·8 per cent. to 25 per cent., an improvement of 33 per cent. on the lower heating value of the gas. The

examination of these and the other tests referred to shows most clearly that indicated thermal efficiency increases with increasing compression, and that the ratio with reference to the air standard in a good, well-designed engine is now from 0·58 to 0·71 of the corresponding air standard efficiency.

It was pointed out by the author many years ago that for equal compression ratios the indicated thermal efficiency increased with increase of engine dimensions. This is clearly shown by the following table from the 1896 edition of this book, and the discussion which accompanied it.

'Comparison of the Actual and Theoretical Efficiencies of Otto Engines of Different Dimensions

Engine cylinder		Relative capacity	Theoretical efficiency	Actual indicated efficiency	Ratio of actual and ideal efficiency
Nearly equal compression	7″ diam. × 15″ stroke .	1	0·428	0·25	$\frac{0·25}{0·428} = 0·58$
	$11\frac{1}{2}$″ diam. × 21″ stroke	3·77	0·428	0·275	$\frac{0·275}{0·428} = 0·64$
Nearly equal compression	$9\frac{1}{2}$″ diam. × 18″ stroke .	1	0·40	0·21	$\frac{0·21}{0·41} = 0·53$
	14″ diam. × 25″ stroke .	2·97	0·41	0·277	$\frac{0·277}{0·41} = 0·67$

From these numbers it is evident that efficiency for equal compression increases considerably with the dimensions of the engine.

'There is, however, a limit to this increase of efficiency with increased dimensions.

'The increase in the efficiency of the larger engines as compared with the smaller using the same proportion of compression space is due to the diminished proportional loss of heat from the gases of the explosion to the inclosing metal walls, and it is always found that in larger engines the expansion curve tends more and more to rise above the adiabatic line. With a maximum temperature of explosion of about 1600° C. it is found by experiment that the actual increase of temperature due to explosion accounts for about from 0·6 to 0·7 of the total heat of the gas present [1]; there is therefore heat enough present in a gas engine of ordinary proportions, if none be lost, to keep up the temperature during expansion performing work to the maximum 1600° during the whole expansion stroke. The increase in dimensions if carried to an extreme could therefore only reduce the loss to insignificant relative proportions, and in such a case the mass of incan-

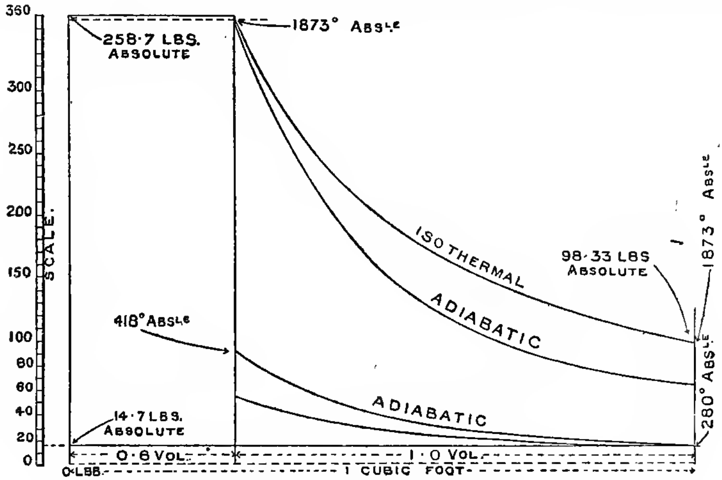
[1] Assuming constant specific heat and air as working fluid.

descent gas might be considered to lose no heat whatever to the walls of the cylinder.

'Assume an air engine in such a case: the total volume of the stroke plus clearance space being 1 cub. ft.

'Assume the engine to have a compression space of 0·275 of the whole cylinder volume, as in the test made by the author on Crossley's Otto scavenging engine. Then the diagram and results would be as shown in fig. 91, where the temperature of explosion is 1600° C.

'From this it will be seen that while 0·409 is the efficiency for adiabatic expansion, then 0·346 is the efficiency for isothermal expansion; from this, then, it appears that, allowing for the known property of the suppression of heat in a gaseous explosion, the utmost efficiency



Efficiency of adiabatic compression and expansion = 0·409.
Efficiency of adiabatic compression and isothermal expansion = 0·346.

FIG. 91.—Theoretical Diagram, comparing adiabatic and isothermal expansion

possible for an engine using coal gas, having a compression space of 0·275 of the total cylinder volume, and expanding to the same volume as existed before compression, is 0·346, so that the efficiency actually attained in practice is $\frac{0{\cdot}277}{0{\cdot}348} = 0{\cdot}80$ or 80 per cent. of the possible.'

In November 1903 a Committee[1] of the Institution of Civil Engineers was formed to consider and report to the Council on the Standards of Efficiency of Internal Combustion Engines. Professor Unwin was the chairman, Prof. Dalby the honorary secretary, and the

[1] Report of the Committee on the Efficiency of Internal Combustion Engines, vol. clxii., session 1904–1905, part iv., and vol. clxiii., session 1905–1906, part i.

other members were Sir Alexander B. W. Kennedy, Professor Callendar, Professor Ashcroft, Captain H. R. Sankey, and Messrs. Hayward, Wilson, and Dugald Clerk. This Committee published a very careful and complete report, recommending the adoption of the air standard; they considered that the standard engine of comparison should satisfy the following conditions:

'(1) The reception and rejection of heat should take place as nearly as may be in the same way as in the actual engine.

'(2) There should be no heat losses due to conduction, radiation, leakage, or imperfect combustion.

'(3) The data for the numerical evaluation of the standard should be ascertainable by simple measurements.

'(4) The expression for the efficiency of the standard should be a simple one.'

Also, that the standard working fluid to operate in the engine be air, which is to be assumed as obeying perfectly the laws of gases. The specific heat is to be assumed as constant for all working temperatures, and the value of γ, the ratio of specific heats at constant pressure and at constant volume, is to be taken as 1·4.

The standard thus recommended is practically that which has been used in all the editions of this book. The members of the Committee were satisfied that, given the ideal efficiencies of a number of engines calculated in this way from the formula

$$E = 1 - \left(\frac{1}{r}\right)^{0.4}$$

that the actual indicated thermal efficiencies obtainable with good gas engines bore a ratio which varied from 0·5 to 0·7. The author has discussed the Committee's report elsewhere[1] as follows:

'The Committee considered the correspondence of the air standard with actual indicated results to be firmly established by these and other tests; but they were not satisfied with the existing knowledge as to the variation of the ratios between the ideal and the actual efficiencies, with varying dimensions of the engines of the same ideal efficiency. It was felt that knowledge was wanting as to the change of ratio, when dealing with engines of widely different dimensions. The author accordingly arranged to place at the Committee's disposal three gas engines of exactly the same type, having the same value of $1/r$ and of quite different dimensions. The cylinders of the engines were respectively 5·5 ins., 9 ins., and 14 ins. in diameter, and the powers were respectively 6, 24, and 60 IHP. All three engines were arranged in one room and supplied with the same gas, and apparatus of the same

[1] "On the Limits of Thermal Efficiency in Internal Combustion Engines," Proceedings, Inst. C.E., vol. clxix. page 157.

type was used for testing them. The tests proved, however, that there was some variation of ratio between the relative efficiencies. Taking the mean mechanical efficiency of the full-load trials as 88 per cent., and calculating the indicated power from the brake power, it was found that the relative efficiencies were respectively 0·61, 0·65, and 0·69. The change of relative efficiency due to change of dimensions was fully proved, and it was clearly shown that by bearing these small changes in mind the engineer could obtain a close approximation to the best actual indicated efficiency to be expected from any given compression used in a gas engine between these widely varying limits. The experiments, in fact, clearly proved the great utility of the air standard for the purpose of comparing different engines with different compression spaces, and predicting the thermal efficiency to be expected from each, assuming the combustion space to be properly proportioned, and the valve actions to be properly performed. The experiments also proved in a quite definite way that which had been long believed by gas-engine builders—namely, the small gain in economy which can be attained by a large engine as compared with a small one. It is somewhat surprising to note, however, that between 6 HP and 60 HP increase of dimensions gives an advantage of only about 12 per cent.

'In recommending the air standard, the Committee were well aware that in proposing air as the standard working fluid with the properties defined in the report, they were using a fluid which differed in its properties from the actual working fluid. If the actual working fluid had the same properties as the ideal, then the relative efficiency numbers given above would truly represent the ratio of the heat converted into work to the heat which could have been converted under the ideal conditions. In the small engine, for example, the relative efficiency number 0·61 would have meant that 0·61 of the heat which the actual working fluid could have converted into work under ideal conditions was converted into work, and with the large engine the proportion was 0·69. That is, the margin to be worked on for improvement was, in one case 39 per cent., and in the other case 31 per cent. It was known from certain investigations that this was not so—in fact, that the possible efficiency of the actual working fluid under ideal conditions was not so high as the number given by the air standard. Had the properties of the actual working fluid been known, it would have been possible to calculate ideal efficiencies using the known properties. Enough was known, however, to show that the actual properties of the working fluid in the engine were by no means simply ascertained, and it was felt better to adopt a standard capable of definite expression, from which the actual best efficiencies could be deduced by a multiplier found experimentally. This multiplier, the relative efficiency, thus includes not only the actual imperfections of the engine cycle, but the

variations between the actual properties of the working fluid and those of the ideal air assumed as a standard.'

In the Committee's trials considerable difficulty was experienced in obtaining the true indicated power by the use of the indicator, although every care was exercised. The difficulty was partly due to variation in the diagram itself when the engine was adjusted to give its best brake efficiency, and partly to imperfections of the indicator itself, which have undoubtedly been accentuated in recent years by the great rise in compression and explosion pressures. The report accordingly states :

'The values for the indicator horse-power, and consequently those for the mechanical efficiency, are probably not very accurate, because the indicator diagrams vary, and the mean of the limited number taken in a trial is not the true mean. It is at least probable that the mechanical efficiency was more nearly constant for the three engines than the figures in the table indicate.'

Further on the report also states :

'It would be desirable, but for one circumstance, to calculate the relative efficiency only from the indicator horse-power. But it appears that in the case of gas engines, and especially gas engines governed by hit-or-miss governors, the indicator diagrams do not give as accurate results as is generally supposed. The diagrams vary much more than those of a steam engine with a steady load, and the mean indicator horse-power, from the diagrams taken in a trial, may, it appears, differ a good deal from the real mean power . . . '

Notwithstanding these difficulties the Committee succeeded in arriving at a satisfactory accurate determination of the mechanical efficiency of each of the three engines by two methods which dispensed with accuracy in the indicator, or, rather, minimised the effect of inaccuracy.

These methods are described in the report as follows :

'The mechanical efficiency values obtained from comparison of IHP and BHP of three Ashton engines are obviously incorrect ; the three full-load tests show :

Engine	L	R	X
Mechanical efficiency . . .	0·90	0·80	0·94

'There appears to be no reason why the R engine should show so low an efficiency as 0·80, and none why L and X should be so high as 0·90 and 0·94. Fortunately the observations made supply means of calculating these efficiencies by two other independent methods :

'(1) By adding IHP without load to BHP at full load, assuming the friction as determined by the indicator to be the same when the engine is running without load as it is when the engine is fully loaded.

'With engine L, Test 4, *No Load* shows that 0·96 IHP maintains the speed of the engine at 291 revolutions per minute; reducing this to 258·9 revolutions per minute, the full-load speed, the IHP necessary to drive the engine without load at 258·9 revolutions is found to be $\frac{258 \cdot 9 \times 0 \cdot 96}{291} = 0 \cdot 85$ IHP.

'The full load at 258·9 revolutions per minute is 5·2 BHP.

$$\text{The IHP is } 5 \cdot 2 + 0 \cdot 85 = 6 \cdot 05$$

and the mechanical efficiency is $\frac{5 \cdot 2}{6 \cdot 05} = 0 \cdot 86$.

'Calculated in this way the respective values of the mechanical efficiency are:

Engine	L	R	X
Mechanical efficiency . .	0·86	0·866	0·888

'(2) By calculating from the full-load and half-load values of the BHP and the gas consumptions, assuming friction to be constant from half to full load.

'The values required are:

BHP at full load.
BHP at half-load.
Gas per hour at full load.
Gas per hour at half-load.

'The BHP and the gas at half-load must be taken at the same number of revolutions per minute as in the full-load trial.

$$\frac{\text{Gas per hour at full load} - \text{Gas per hour at half-load}}{\text{BHP at full load} - \text{BHP at half-load}} = \text{Gas per IHP hour}$$

$$\text{Mechanical efficiency} = \frac{\text{Gas per IHP hour}}{\text{Gas per BHP hour}}$$

'Calculated in this way the mechanical efficiencies are:

Engine	L	R	X
Mechanical efficiency . . .	0·81	0·83	0·84

'The mean values by (1) or (2) are:

Engine	L	R	X
Mechanical efficiency . .	0·835	0·848	0·864

'Method (1) depends on the accuracy of the indicator; but an error of, say, 5 per cent. only introduces an error of that amount in the friction value itself. In calculating mechanical efficiency from the total indicated power an error of 5 per cent. on the total may readily amount to 20 per cent. on the friction, while by method (1) it is limited

to the 5 per cent. on the friction value calculated. Method (2) gives the mechanical efficiency without reference to the indicator, and it only assumes that the diagrams remain constant at the lighter load and that friction is constant between full and half-loads.

'It seems clear, as has already been stated, that however carefully indicator diagrams are taken in gas-engine trials, they do not furnish as accurate a value of the mean indicator horse-power as has been generally supposed.'

If, then, the values 0·84, 0·85, and 0·86 be taken as the mechanical efficiency of the three engines 'L,' 'R,' and 'X,' a very close approximation to the truth will be obtained.

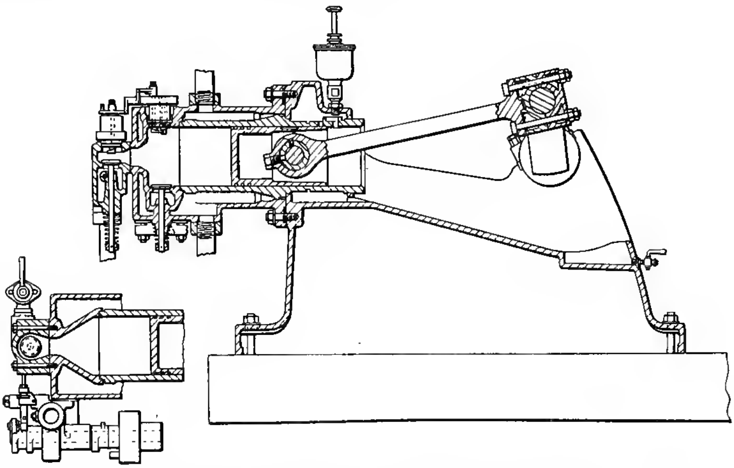


FIG. 92.—Sections of 'L' Engine. (National Gas Engine Co., Ltd.) Institution of Civil Engineers' Committee Tests

The brake thermal efficiencies of the Committee, which were undoubtedly accurate, were as follows :

Engine	L	R	X
Brake thermal efficiency, per cent.	26·1	28	29·9

Dividing these values by 0·84, 0·85, and 0·86 respectively, we get :

Engine	L	R	X
Indicated thermal efficiency, per cent.	31	32·9	34·8

The air standard efficiency for the three engines is :

Engine	L	R	X
Air standard efficiency, per cent.	49·6	49·6	49

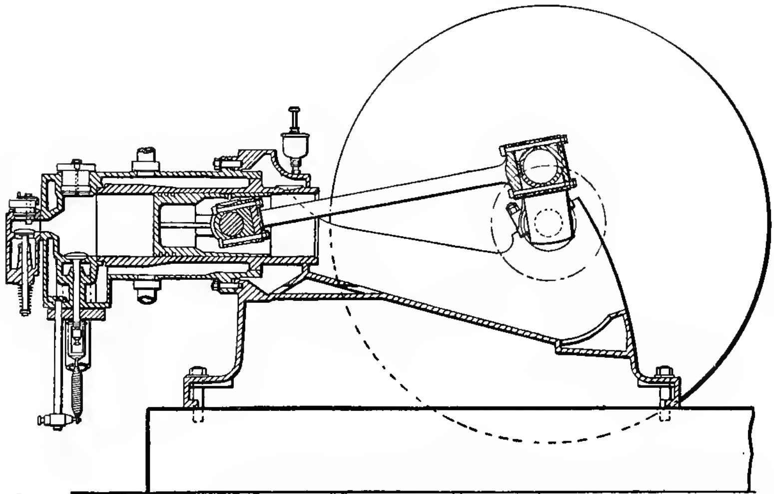
Dividing by the respective indicated thermal efficiencies, we get :

Engine	L	R	X
Relative indicated efficiency	0·625	0·662	0·71

It may be taken as established, then, from the experiments of the Committee, that when three engines of different dimensions are adjusted to give their best economical result in gas consumption per brake horse-power, that the efficiency ratio relative to the air standard varies from the value 0·62 to 0·71, the cylinders being respectively 5·5 ins., 9 ins., and 14 ins. diameter.

The conclusions thus support those of the author already mentioned.

As these experiments of the Institution of Civil Engineers are of great importance, it is desirable to describe the engines experimented



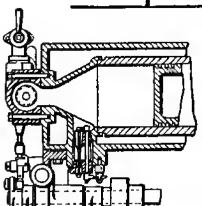
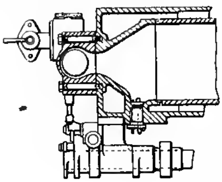


FIG. 93.—Sections of 'R' Engine. (National Gas Engine Co., Ltd.)
Institution of Civil Engineers' Committee Tests

with and some of the other results obtained. The three engines designated L, R, and X are shown in longitudinal section and part horizontal section at figs. 92, 93, and 94. Some particulars are marked under the figures. They are described as follows in the Committee's report:

'The three engines tested were of the standard four-stroke cycle type, built by the National Gas Engine Co., Ltd., for cylinders of 5·5, 9, and 14 ins. diameter respectively. The arrangements of the engines were identical in respect of the proportions of the combustion space, the mechanism for the admission of the charge, and the exhaust valves. In each case the charge was admitted by means of an inlet valve opening into a central port at the end of the combustion

space. Behind the inlet valve was a gas valve, and within the inlet port the electric igniter was arranged, operating on the low-tension principle, with a Simms-Bosch magneto-instrument. The three engines were each provided with two electrical igniting arrangements, the L engine having the second igniter placed within the cylinder in a plug above the exhaust valve. The R and X type engines had the second igniter placed at the side of the combustion space, and not at the top. These second igniters were introduced, as it was intended at one time to make experiments with ignition inside the cylinder instead of in the port. These experiments, however, were not made by the Committee. The inlet valve in each case was operated from the cam on the usual two-to-one shaft, which runs along the side of the cylinder, and the gas valve was operated by a similar cam placed close



to the main supply-valve cam. The engines were governed on the hit-or-miss method by the action of a centrifugal governor on a small block, which engaged with a pecker. The exhaust valves were placed in the bottom of the combustion space, and were operated

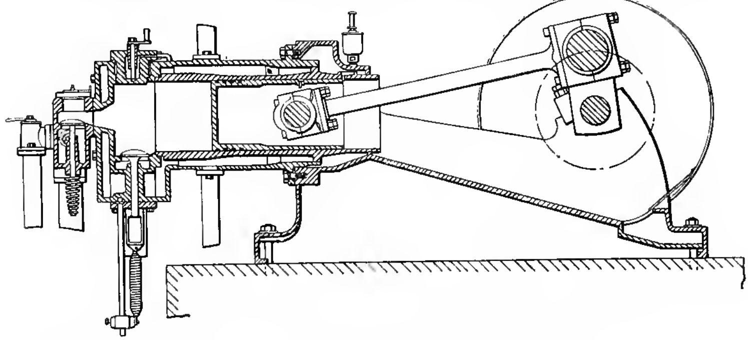


FIG. 94.—Sections of 'X' Engine. (National Gas Engine Co., Ltd.)
Institution of Civil Engineers' Committee Tests

also from the two-to-one shaft by levers. The exhaust gases, after passing the exhaust valve, traversed a water-cooled passage, the water-jacket of which was included in the jacket circulation of the engine. This is clearly seen in the section of the exhaust calorimeter, fig. 95. It will be noticed that with this arrangement heat is abstracted from the exhaust gases after opening the exhaust valve, before these gases arrive at the exhaust calorimeter.

'In the large engine, that is, the X type, the starting was accomplished by means of a hand-pressure pump, which communicated with a plug above the exhaust valve. This plug contained a valve which

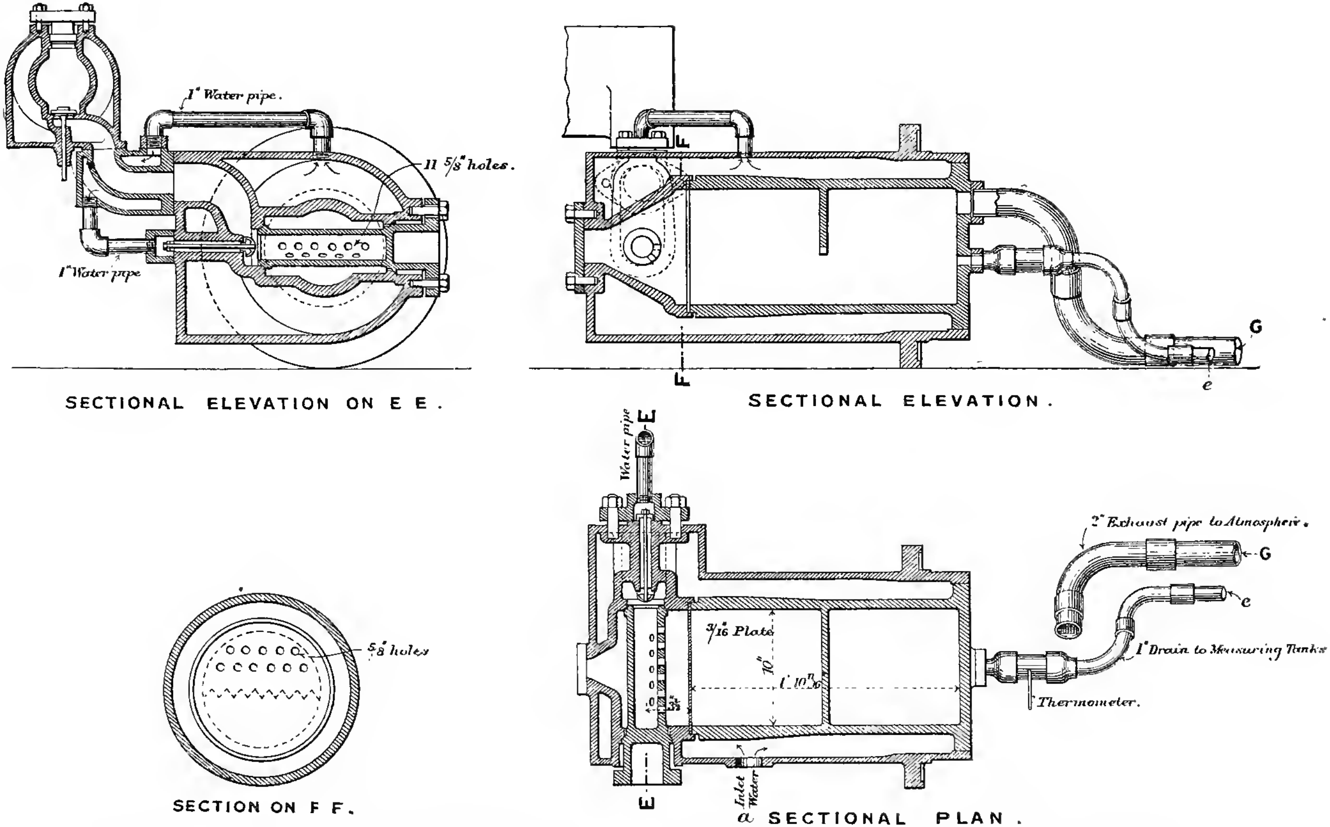


FIG. 95.—Exhaust Gas Calorimeter as coupled to Engines 'L,' 'R,' and 'X.' Institution of Civil Engineers' Committee Tests

(*Reproduced by the kind permission of the Institution of Civil Engineers*)

allowed the access of mixture to the cylinder from the hand-pump. The engine was started by placing the crank conveniently over the centre, pumping in a mixture of gas and air behind the piston, to a pressure slightly above that of the atmosphere, shutting the inlet valve in the plug, and tripping the magneto-shield of the Simms-Bosch instrument. The electric spark then passed between the separated surfaces within the cylinder, and the engine started.

'The Simms-Bosch magneto was of a well-known type, having a fixed armature, fixed permanent magnets, and movable shield. The shield was withdrawn and tripped by the operation of an adjustable pin rotating on a collar on the two-to-one shaft. This pin can be adjusted to vary the time of firing the charge as may be required.

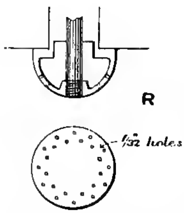


Fig. 96.—Details of spray of Exhaust Gas Calorimeter of fig. 95

'It is needless to describe the operation of the engines. They act on the ordinary four-stroke cycle, and there was nothing in the mechanical arrangements which calls for special description. They are the well-known arrangements of the National Gas Engine Co., as applied to engines of the three sizes selected by the Committee.

'It may be stated, however, that in order to secure results comparable scientifically, the compression spaces were carefully adjusted to have as nearly as possible similar proportions in the three engines. Care, too, was taken that the internal surface of the combustion spaces should be smooth and clean, in order that no unknown element should enter into the observations due to irregularities in the castings. In the small engine, care was taken that the working parts were all very free, in order that no undue friction should affect the results.

Dimensions of Engines

(*Captain Sankey and Professor Dalby*)

Designation of engine	L	R	X
Clearance volume in cubic centimetres . . .	850	3,920	12,680
Clearance volume in cubic inches (1 cubic inch = 16·387 cubic centimetres) . . .	52	239	774
Diameter of cylinder . . . inches	5·502	9·00	14·008
Stroke . . . . ,,	10·00	17·03	22·00
Area of cylinder . . . . . . square inches	23·78	63·62	154·1
Volume displaced by piston-stroke . cubic inches	237·8	1,083	3,390
Total volume of cylinder . . ,, ,,	289·8	1,322	4,164
Clearance. Percentage of total volume . . .	17·94	18·08	18·59
Circumference of brake-drum . . . . . feet	9·19	18·208	19·84
,, ,, rope . . . . . ,,	0·01	0·230	0·295
Effective circumference of brake . . . . ,,	9·200	18·438	20·141
Diameter of air-orifice in the measuring trunk, inches	4	8	12

'The three engines were adjusted as to gas-supply so that each engine should run as nearly as possible at its most economical load.

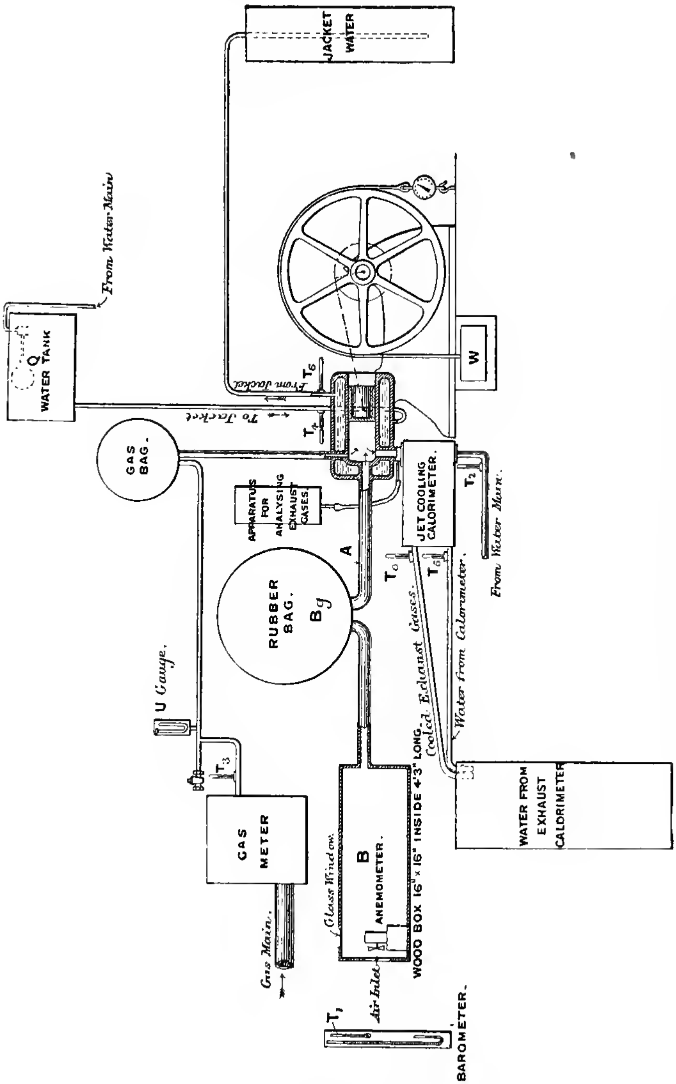


FIG. 97.—Diagrammatic arrangement of engine and apparatus. Institution of Civil Engineers' Committee Tests.

(*Reproduced by the kind permission of the Institution of Civil Engineers*)

With more gas the three engines could each give considerably more power, but such power would partake somewhat of the nature of an overload, and the consumption per brake HP would not be quite so

low. In all the tests made, the igniter used was that in the admission port.'

Fig. 97 shows a diagrammatic arrangement of the apparatus used

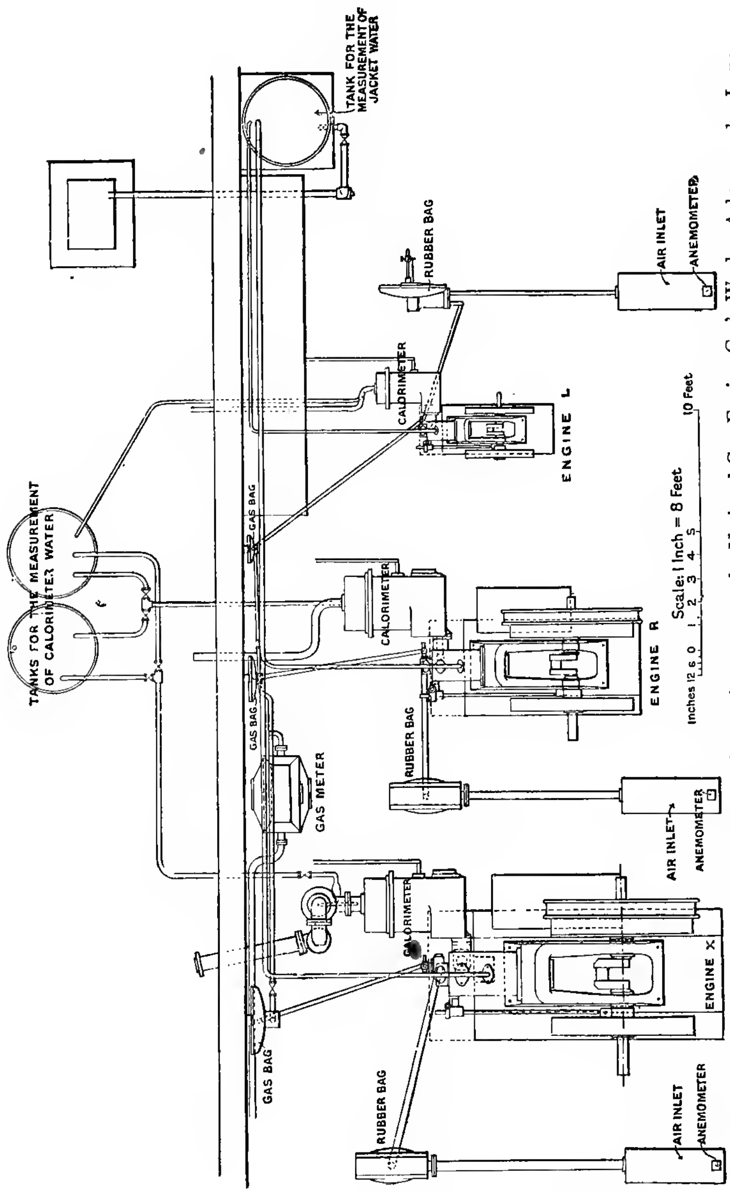


FIG. 98.—General arrangement of the engine testing-room at the National Gas Engine Co.'s Works, Ashton-under-Lyne. Institution of Civil Engineers' Committee Tests.

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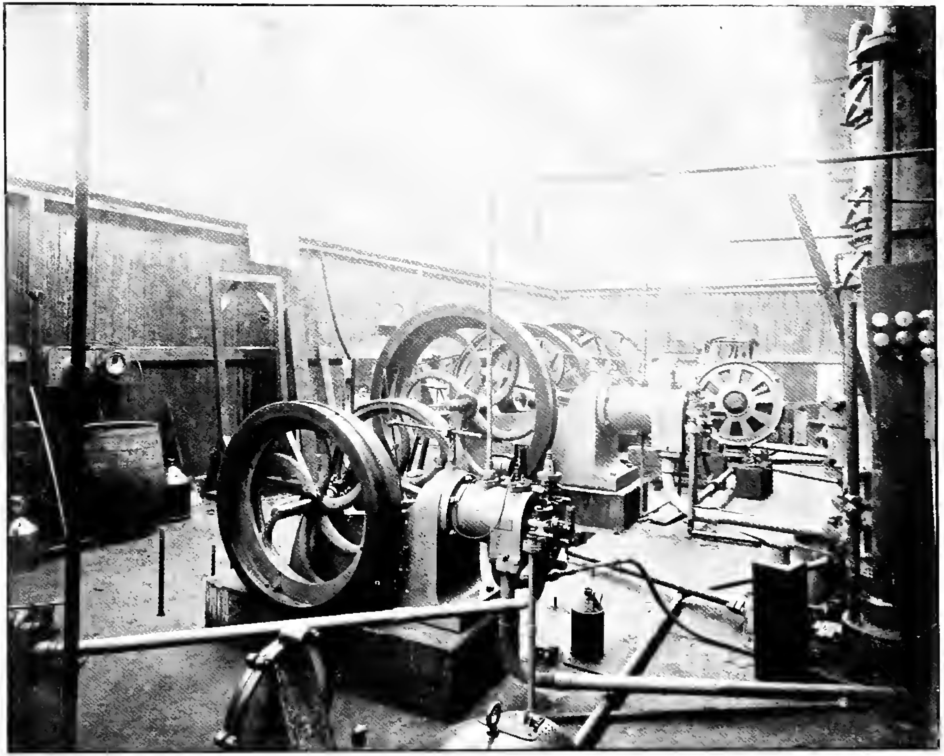

Fig. 99.—Engine testing-room at the National Gas Engine Co.'s Works, Ashton-under Lyne, with engines 'L,' 'R,' and 'X' in position.

Institution of Civil Engineers' Committee Tests

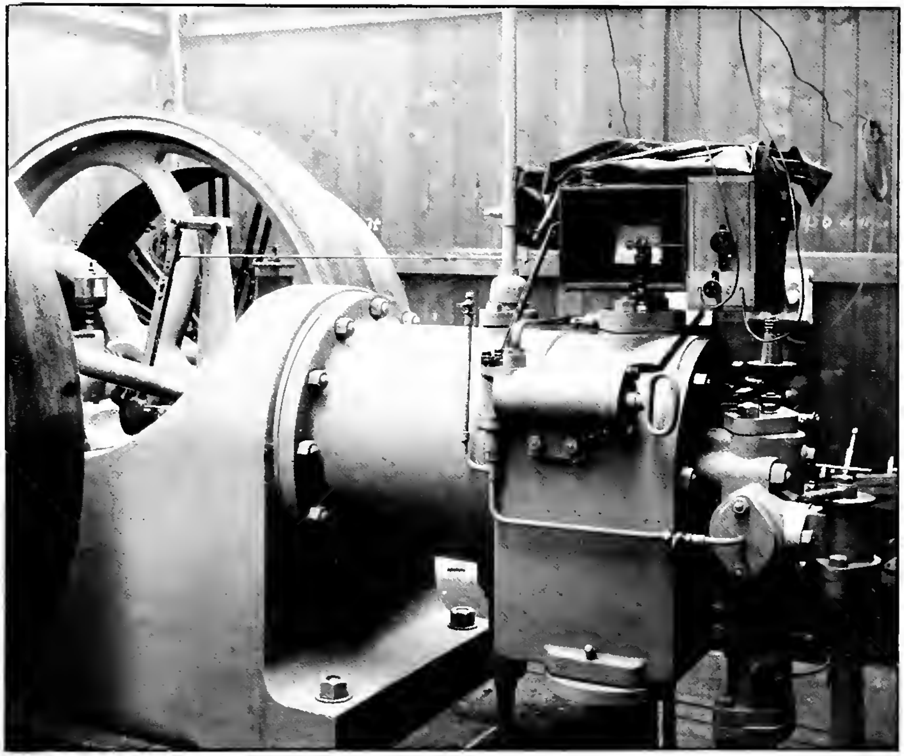

FIG. 100. 'X' engine with a Clerk optical indicator used for Clerk diagrams as described in Chapter VIII

in the testing of each engine. A novel feature in connection with this is the method adopted for measuring the air-supply to the engine by means of an anemometer placed in a box which forms an enlarged continuation of the air-suction pipe. A novel form of exhaust-gas calorimeter, designed and fitted by the Company, was used to extract the heat from the exhaust gases; this was arranged on the principle first used by Professor Bertram Hopkinson, and is given at figs. 95 and 96. The water measurements were made in tanks, which were calibrated by the Committee. All the weights, spring balances, and thermometers used were calibrated.

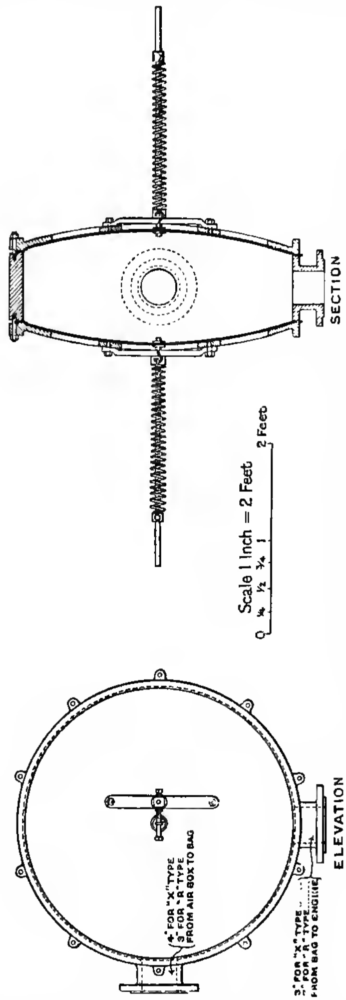

Fig. 101.—Air pressure equalising-bag between engine and anemometer-box

Fig. 98 is a plan showing the general arrangement of the engine-room, and fig. 99 is a photograph of the room with the engines in position.

Fig. 100 is a photograph of part of the large or 'X' engine, showing an optical indicator of the author's design as arranged for experiments such as those indicated in an earlier chapter.

The report proceeds:

'The measurement of air quantity was made by means of a small anemometer, placed inside a wood trunk B, fig. 97, which was attached to an extension of the air-suction pipe A of the engine. A large gas-bag, the details of which are shown in fig. 101, was placed on the connecting pipe in order to keep the flow of air at the anemometer as uniform in speed as possible. The fan of the anemometer was placed close to the air inlet to the box, and its indications were read from without through a glass window let into the top of the box. The air inlet was circular, and was cut out of a piece of sheet-iron so that the size could be easily adjusted to determine a rate of flow past the anemometer which

would give a suitable rate of rotation to the fan. The pressure and temperature of the entering air were measured by a barometer and a thermometer T_1, whilst the hygrometric condition was taken by means of a wet- and dry-bulb thermometer.

'At the engine trials, readings of the anemometer were observed which gave the number of lineal feet of air passing the anemometer. These had to be reduced by some means to cubic feet of air. The method devised for doing this was simple and effective. When the engine was quite cold, it was driven by an electric motor through a belt placed on the flywheel. The gas-supply was cut off and the engine

ANEMOMETER CALIBRATING TRIALS

Number of test	Diameter of orifice	Anemometer reading. Velocity of air	Revolutions of engine per minute	Volume displaced by piston per stroke	Ratio of effective to total stroke	Volume of air through orifice per minute	Volume of air corresponding to anemometer unit
	Ins.	Ft. per min.		Cub. ft.		Cub. ft.	Cub. ft.
5	4	293·3	285·0	0·1376	0·886	17·26	0·0588
6	,,	268·0	263·0	,,	0·920	16·65	0·0621
						Mean	0·0605
11	8	227·3	214·1	0·6273	0·892	59·9	0·263 †
13	,,	118·2	142·5	,,	0·756	33·8	0·285
1 *	,,	185·5	84·8	,,	1·000	53·2	0·287
2 *	,,	268·5	125·5	,,	1·000	78·7	0·293
3 *	,,	357·5	163·0	,,	0·977	99·9	0·279
						Mean	0·281
18	12	193·8	142·2	1·962	0·927	129·3	0·667
19	,,	214·3	155·6	,,	1·000	152·6	0·712
4 *	,,	138·0	165·0	,,	0·988	102·3	0·741
5 *	,,	166·0	193·0	,,	0·977	118·3	0·713
						Mean	0·708

* These tests were made with special cams for the valve-gear so that the engine acted as a suction pump without compression. † This result is rather anomalous.

allowed to draw in air through the orifice used in the trial, the cylinder in this way serving as a calibrating chamber. In some subsequent tests made by Mr. Dugald Clerk on the engine 'R,' the cams were changed so that the cylinder became a single-acting air-pump without compression, and two of the orifices were separately calibrated on this engine at the mean speed of the actual trial. The dimensions of the box are given in fig. 97.

'In these anemometer calibrating trials the number of revolutions of the engine, the anemometer readings, and the duration of the run were observed. Indicator diagrams also were taken. It was assumed

that the proportion of the cylinder filled with air at atmospheric pressure at each stroke was given by the length on the indicator diagram between the points at which the pencil-line crossed the atmospheric line. This distance may be termed the effective stroke. From these results the cubic feet of air per unit reading of the anemometer for each orifice used were obtained.

'The hot gases were cooled by passing them through an exhaust-gas calorimeter. The construction of the calorimeters used in the trials is shown by the working drawings, figs. 95 and 96. It will be observed that the calorimeter is water-jacketed right up to the connection with the engine exhaust flange. The water is led in at *a*, and after passing through the jackets is led to the rose, R, fig. 96, through the small orifices of which it spurts out to meet the stream of exhaust gas. The water and gas then find their way out, passing various obstructions to ensure mixing and abstraction of heat from the gas, until finally the gas, cooled down to about 90° F., escapes at G, and the water escapes at *e*, having about the same temperature as the escaping gas, although sometimes it was higher and sometimes lower. The general arrangement of the calorimeter is shown in fig. 97, from which it will be seen that the water was brought into the calorimeter directly from the water main, and was led to measuring tanks placed outside the testing-room. These tanks were fitted with gauge glasses and scales graduated in feet and decimals of a foot, and were calibrated by pouring in weighed quantities of water. The calibrations obtained were:

	Lbs. of water per ft.
No. 1 tank for exhaust calorimeter	680·82
No. 2 ,, ,, ,, ,, . .	679·42

'In the design of these calorimeters care should be taken that the thermometer measuring the temperature of the escaping cooling water is placed so that the bulb is completely immersed in the water. As the pipe does not always run full, a pocket should be formed in the pipe to ensure this condition. Care should also be taken that the gas itself does not carry away water mechanically suspended in it. The drawings, figs. 102, 103, and 104, show a form of calorimeter in which special attention is given to these two points, and which was used in some trials made by Mr. Dugald Clerk subsequent to those made by the Committee. The results showed that, although the calorimeters used by the Committee were not provided with these safeguards, the quantities concerned were measured without appreciable error.

'In the reduction of the observations on the calorimeter, several minor points have to be observed. In the first place, the water flowing out of the calorimeter, and which is measured in the measuring tanks,

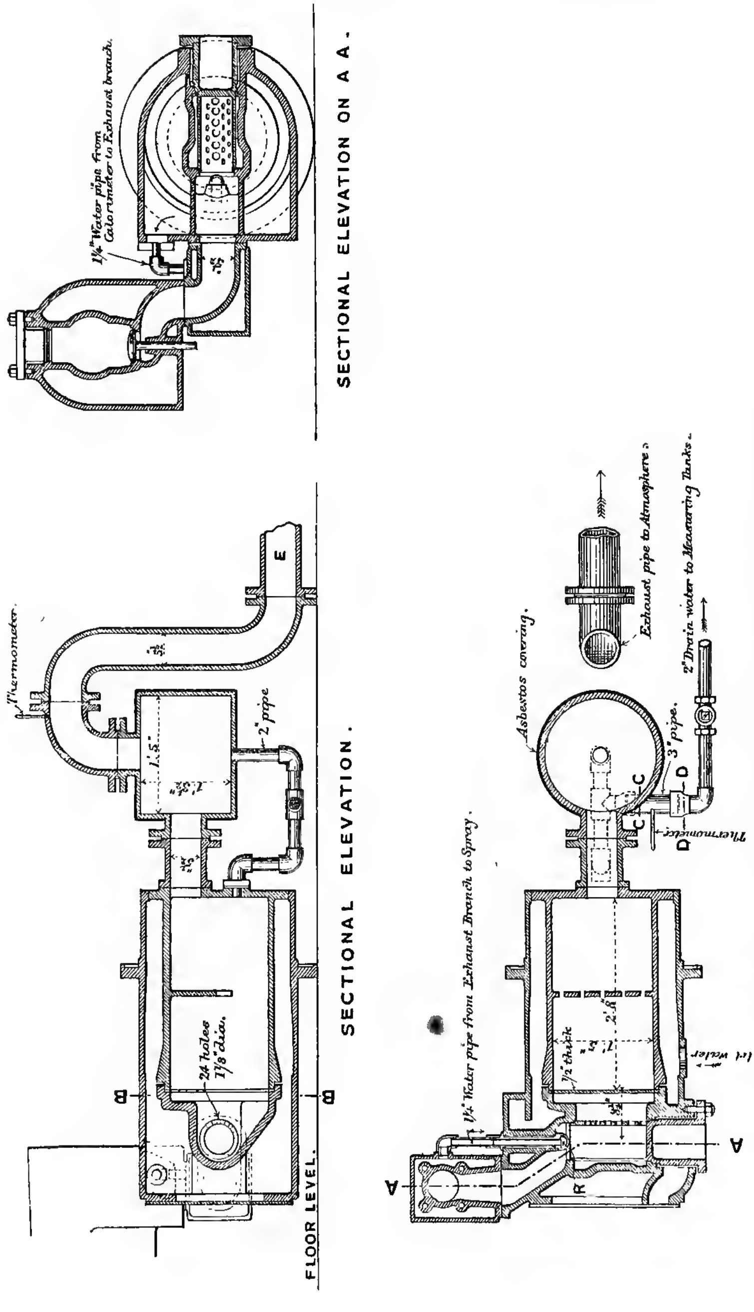

FIG. 102.—Modified Exhaust Gas Calorimeter recommended by the Inst. C.E. Committee, experimented with by Dugald Clerk.
(Reproduced by the kind permission of the Institution of Civil Engineers)

is not exactly the quantity which enters it. The gases cooled to about 90° F. are at this temperature like a sponge as regards the absorption of water-vapour, and as they have been in intimate contact with the water during the whole time of the passage through the calorimeter it may be assumed that they will pass out completely saturated with moisture at the exhaust temperature. Now the amount of water required to saturate the exhaust gases may be computed as though the gas was entirely air without introducing serious error. . . .

'This amount is not, however, all abstracted from the water entering the calorimeter, for it will be observed that the air entering the engine cylinder brings in with it a definite weight of moisture, and the combustion of the gas produces a definite weight also. Hence the quantity actually abstracted from the cooling water is the difference between the quantity required for the saturation of the exhaust gases

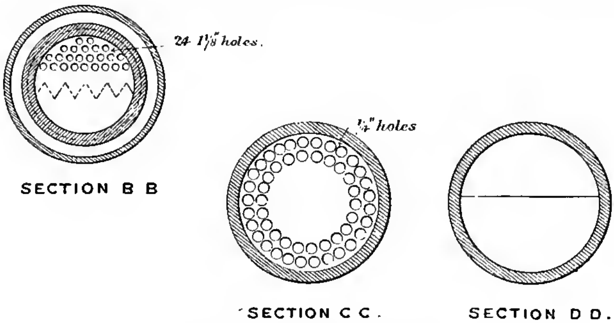

Fig. 103. – Details of fig. 102

and the sum of the water produced by combustion and that brought in by the air.'

Radiation of heat from the engine surface is difficult to determine, and it is dealt with as follows :

'The Committee include under the heading of radiation the radiation from the hot surfaces of the cylinder, trunk piston, &c., and the radiation from the bearings of the engine due to friction. In other words, the term includes the heat lost by direct radiation from the hot surfaces, together with the heat corresponding to the difference between the indicator horse-power and the brake horse-power. The difficulty of finding the indicator horse-power exactly led the Committee to define radiation in this way.

'The method of measuring this quantity was to run the engine light and then to adjust the number of explosions per minute so that the temperatures maintained on the thermometers measuring the temperatures of the water to and from the jackets were respectively equal to those at a full-load trial. In this way the surface temperatures,

allowing a very small quantity of water to flow through the jackets in order to obtain similar conditions to those during the normal working of the engine, were kept approximately the same in the two trials, although the inner temperatures would be lower because of the fewer explosions.

'The method is of course only an approximate one, but it gives some idea of the losses from this cause.'

Very full and interesting details of measurements and corrections are given in the report, and the reader is referred to it for further study; but some of the most generally interesting results are given in the following table:

Engine	L	R	X
Brake HP	5·2	20·9	52·7
Revs. per minute	258·9	203·6	165·8
Cyl. diameter and stroke	5″·5 × 10″	9″ × 17″	14″ × 22″
Gas per BHP hour at working temp. and pressure	16·87 cub. ft.	15·84 cub. ft.	14·9 cub. ft.
Lower calorific value of gas at working temp. and pressure	566 B.Th.U.	567 B.Th.U.	574 B.Th.U.
Ratio by volume of air to gas in charge	$\frac{\text{air}}{\text{gas}} = \frac{9·15}{1}$	$\frac{\text{air}}{\text{gas}} = \frac{9·17}{1}$	$\frac{\text{air}}{\text{gas}} = \frac{8·21}{1}$
Ratio by volume of air + exhaust to gas in charge	$\frac{\text{air + exhaust}}{\text{gas}} = \frac{10·1}{1}$	$\frac{\text{air + exhaust}}{\text{gas}} = \frac{9·75}{1}$	$\frac{\text{air + exhaust}}{\text{gas}} = \frac{9·3}{1}$

It is to be noted that the brake power given was found to give the most economical result per brake horse in each engine, and it will be seen that the proportion of air to gas in the L and R tests was practically as 9·15 to 1, while the 'X' engine used the somewhat stronger mixture of 8·21 to 1 air to gas. The author has calculated the total dilution in the case of the 'X' engine as $\frac{\text{air + exhaust gases}}{\text{gas}} = \frac{9·3}{1}$.

It will be observed that the Committee determined by direct methods all the values necessary to enable a complete balance-sheet of the disposal of heat to be prepared for each engine.

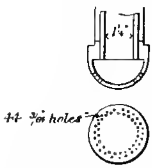

Fig. 104.—Detail of Nozzle, fig. 102

The coal gas used was measured by meter, its heating value was determined by Junker calorimeter, and it was also analysed chemically. This enables the total heat given to each engine in the form of chemical energy to be accurately known.

The heat leaving the engine by way of the hot exhaust gases was accurately known from the readings of the exhaust calorimeter.

The heat leaving by the water jackets was also accurately determined.

Total radiation from the engine was also determined, but here the method used was rough, and the total radiation value cannot be considered as better than a fair approximation.

The brake power of the engine was very accurately determined; the values for brake power are the most reliable of all.

By adding together all the values so determined, it should be possible to account for all the heat given to each engine.

The following balance-sheets are given for the full-load tests in the report:

INSTITUTION OF CIVIL ENGINEERS' COMMITTEE TESTS—HEAT BALANCE-SHEET. FULL LOAD

Designation of engine . . .	L	R	X
Exhaust waste	35·3	40·0	39·5
Jacket waste	23·5	29·3	25·0
Radiation	7·6	10·0	7·3
BHP	26·7	28·3	29·8
	93·1	107·6	101·6

Although every care was exercised, the experiments do not account for all the heat given to the small or 'L' engine; of 100 heat units given to the engine only 93·1 units have been found—a loss of 6·9 per cent. has been incurred in some way. Analysis of the exhaust gases was made to find if all the gas had been burned and combustion appeared to be complete, so that the deficit was not accounted for in this way.

In the tests of the larger engines 'R' and 'X' too much heat has been found by 7·6 per cent. on 'R' and 1·6 per cent. on the 'X' engine.

The test of the large engine comes most nearly to the 100 heat units, and this greater accuracy would be expected as the heat quantities measured were greater and error more easily avoided.

By reasoning upon other figures found in the report the author has discussed this balance-sheet, and has arrived at the conclusion that the following adjusted balance-sheet more truly represents the actuai disposition of heat given to the three engines:

INSTITUTION OF CIVIL ENGINEERS' COMMITTEE TESTS—HEAT BALANCE-SHEET OF THE THREE ENGINES ADJUSTED BY CLERK

Designation of engine . . .	L	R	X
Exhaust waste . . .	34·1	37·1	39·9
Jacket waste and radiation .	34·1	29·6	25·4
IHP . . .	31·8	33·3	34·7
	100·0	100·0	100·0

Here the indicated power is given instead of the brake power, so that the friction of the engine is no longer included under radiation.

The reasons for this adjustment will be found in the paper already referred to, read by the author before the Institution of Civil Engineers in 1907; they need not be dealt with here.

In all gas-engine balance-sheets as hitherto presented the jacket loss is over-estimated. The jacket item in the balance-sheet should represent the heat flow from the hot gases in the engine cylinder during explosion and expansion, but a considerable proportion of heat finds its way into the jacket water after expansion is complete; that is, heat comes from the exhaust gases upon discharge, which should appear under exhaust waste.

The adjusted balance-sheet above is erroneous in this respect: too much heat appears under jacket waste and radiation, and too little under exhaust waste. There are several reasons for this. One is that when the exhaust valve opens, the hot gases discharging round the valve impinge violently upon the water-jacketed metal beneath the valve before their course is turned, and then pass through a water-jacketed space before reaching the water-jacketed passage included in the calorimeter space. As gases in violent motion impinging against a metal surface lose heat very rapidly, it follows that some of the heat which should appear in the exhaust calorimeter appears in the water jacket. Further, when the gases in the cylinder have fallen to atmospheric pressure, the cylinder still remains filled with hot gases, often at a temperature of over 1000° C. During the exhaust stroke of the engine piston these gases are in contact with the sides of the cylinder, and are flowing through the combustion space and valves, so that a good deal of remaining heat also passes into the water jacket. The friction of the piston, too, generates heat, and this heat either flows into the water jacket or disappears in radiation and conduction from the piston-bottom to the external atmosphere.

This difficulty cannot be effectively met by any of the ordinary methods hitherto used, and for the purpose of arriving at a more accurate division the author has continued experiments upon the large 'X' engine of the Committee by his new diagram method described in the last chapter.

The work of the Institution of Civil Engineers' Committee has proved beyond doubt that the air standard furnishes a reliable measure of the varying efficiency of different engines; that the actual indicated thermal efficiency may be readily deduced from the theoretic thermal efficiency by the use of a multiplier which varies to a small extent with the dimensions of the engine; that in the best modern engines of varying dimensions this multiplier varies between 0·62 and 0·71; that the ordinary indicator does not give so reliable a result as is commonly supposed; that it is more difficult to obtain accurate indicated power from the gas engine than is usually assumed; that this difficulty

may be overcome by combining in use the brake and indicator for light loads ; that the anemometer may be used to determine air-supply to the engine ; that radiation may be approximately determined by the new method described, and, finally, that in the three engines tested brake efficiencies of 26·7, 28·3, and 29·9 per cent. are readily obtained, corresponding to indicated thermal efficiencies of 31·8, 33·3, and 34·7 per cent. These appear to be the highest thermal efficiencies obtained and authoritatively vouched for up to the date of the Committee trials.

These results are of the utmost importance and supply a distinct step in the development of the science of the internal-combustion motor.

It has been pointed out in the preceding discussion that the air standard efficiency is higher than the efficiency which could be obtained from the actual working fluid, knowing its properties as we now know them.

In the preceding chapter the author has described a new method of determining the cooling curve in the actual operation of the piston in the gas-engine cylinder, and has shown that not only can a cooling curve be deduced giving the temperature fall due to cooling to the walls, but the dynamic value of this fall can be deduced in foot-pounds per cubic foot of working fluid. It will be at once recognised that this method supplies a means of arriving at a balance-sheet of the gas engine by indicator measurements only; it does not require any gas measurements or heat-flow measurements whatever ; it only requires diagrams of the type shown at p. 268 to be able to arrive at a balance-sheet without the immense labour of the usual methods of testing.

A test of this kind was made by the author on the 'X' engine of the Institution of Civil Engineers' Committee report at the brake load of 50 HP. Three diagrams were taken in the manner described in Chapter VIII.

These diagrams are given at fig. 105.

The mean temperature in terms of time was determined for each explosion and expansion stroke ; the temperature fall due to cooling to the walls was taken, and the dynamic value of the temperature fall was obtained from the table of apparent specific heats given at p. 235. The temperature of the gases at the end of the expansion was measured by the pressure at that point with the piston in the full-out position. From this temperature and the initial temperature of the charge the heat carried away by the exhaust gases was determined by using the apparent specific-heat table referred to. Then the indicated work was obtained from the positive loop of the diagram—that is, the area between the compression line and the explosion and expansion line.

The volume of air entering the engine was determined by anemometer and the gas by gas meter, the temperature of the external air

was also taken, and the barometric pressure. The temperature of the water jacket was the usual full-load temperature of 80° C.

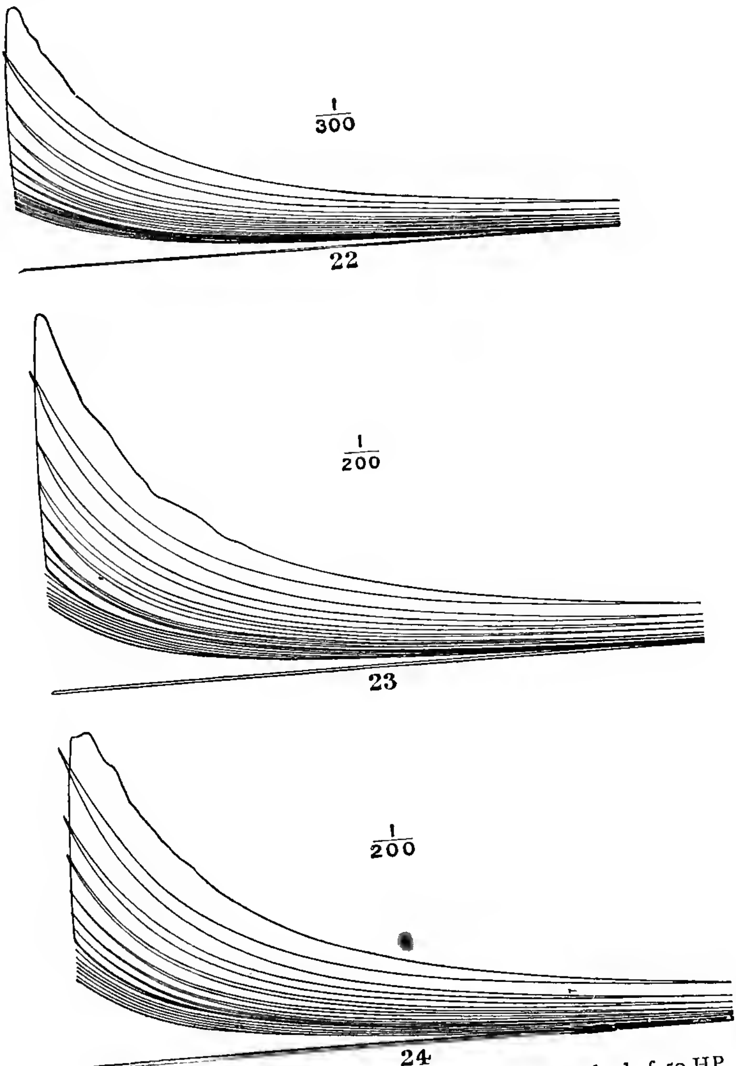

FIG. 105.—Clerk diagrams from 'X' engine at a brake-load of 50 HP at 160 revolutions per minute

In this way the following three balance-sheets were prepared for the three indicator-cards Nos. 22, 23, and 24.

HEAT BALANCE-SHEET FOR 'X' ENGINE PREPARED FROM CLERK DIAGRAMS, FIG. 105. (*Clerk*)

Card No. 22

	Ft.-lbs.	Per cent.
Heat flow during explosion and expansion	12,480	15·4
Heat contained in gases at end of expansion	39,800	49·0
Indicated work	28,900	35·6
Total heat	81,180	100·0
i.e., 104 B.Th.U.		

Card No. 23

	Ft.-lbs.	Per cent.
Heat flow during explosion and expansion	14,000	17·0
Heat contained in gases at end of expansion	40,500	49·3
Indicated work	27,700	33·7
Total heat	82,200	100·0
i.e., 106 B.Th.U.		

Card No. 24

	Ft.-lbs.	Per cent.
Heat flow during explosion and expansion	13,100	16·0
Heat contained in gases at end of expansion	40,600	49·5
Indicated work	28,260	34·5
Total heat	81,960	100·0
i.e., 106 B.Th.U.		

All the values have been taken in foot-pounds. Taking card No. 22, for example, the temperature fall due to cooling is equal to 12,480 ft.-lbs., the heat contained in the gases at the end of the expansion is equal to 39,800 ft.-lbs., and the positive loop of the indicator diagram shows that the work done on the piston is 28,900 ft.-lbs.

Now, if we add together these three items we should get the total heat given to the charge for one stroke of the engine; this amounts, it would seem, to 81,180 ft.-lbs., which is equivalent to 104 British thermal units.

Cards 23 and 24 vary slightly from this, but they show each 106 B.Th.U.

Now, if the combustion is nearly complete at the end of the stroke the heat present found in this way should be equal to the heat evolved by the gas known to be present in the charge. The gas present in the charge is known to be approximately 0·183 cub. ft. at the working temperature of the measuring meter, and its lower calorific value was 574 B.Th.U. per cubic foot.

The heat of combustion of the gas is therefore

$$0{\cdot}183 \times 574 = 105 \text{ B.Th.U.}$$

It is thus seen that the approximation is very close. The indicator

has been able by the new method of application to account for the heat present in the charge.

The distribution of the heat varies to a small extent in the three diagrams, so that it is better to consider the mean result.

The mean balance-sheet of these three from cards Nos. 22, 23, and 24 is as follows:

MEAN BALANCE-SHEET, CARDS NOS. 22, 23, AND 24

	Per cent.
Heat flow during explosion and expansion	16·1
Heat contained in gases at end of expansion	49·3
Indicated work	34·6
	100·0

These indicator diagrams, together with the apparent specific heat values given in Chapter VIII., thus enable the total heat evolved in the cylinder per stroke, and its distribution in indicated work, necessary exhaust loss, and heat flow through the cylinder walls, to be determined from the diagram only.

Compare this balance-sheet with that deduced from the Committee's trials on p. 265.

—	Committee's trials	New diagram trials
Heat flow during explosion and expansion	25·4	16·1
Heat contained in gases at end of expansion	39·9	49·3
Indicated work	34·7	34·6
	100·0	100·0

The indicated work is practically the same in both trials and the sum of the other two items is the same also, but the distribution is different. Less heat flows through the cylinder walls as determined by the author's new method, and the exhaust gases contain more heat than the Committee's calorimeter trials show. The ordinary trials show 9·3 per cent. too much heat as passing through the cylinder-walls, and practically the same amount too little appears in the exhaust calorimeter. That is, 18·8 per cent. of the total heat remaining in the hot gases at the end of the expansion passes into the cylinder water-jacket during the flow through the exhaust valve upon the first opening and while the piston is making its exhaust stroke.

Or, roughly, the true heat flow on explosion and expansion is about 0·63 of that usually measured by water jacket and radiation. This seems to be a quite reasonable portion of the total heat, such a portion as experience would lead one to expect.

These new diagram trials afford, in the author's view, a more accurate heat distribution balance-sheet than has yet been obtained

in any engine, from which can be deduced the ideal efficiency of the working fluid.

If it be assumed that in this experiment the whole of the heat loss, say 16 per cent., is incurred at the beginning of the stroke before the attainment of maximum temperature, then the total heat dealt with during expansion would be 100 — 16 = 84, and the thermal efficiency had there been no loss would have been $\frac{34 \cdot 6}{84} = 0 \cdot 41$.

That is, the best efficiency which this working fluid could give for the particular compression ratio $\frac{1}{r}$ would be 41 per cent. Had air been the working fluid, it has been already pointed out, the ideal efficiency would be 49 per cent., so that the possible efficiency from the actual working fluid is considerably less than that possible from air.

The heat loss is not all incurred at the beginning of the stroke. It has been shown in the last chapter that about 50 per cent. of the total heat loss is incurred during the first $\frac{3}{10}$ of the stroke, and the remaining 50 per cent. in the last $\frac{7}{10}$. Even 41 per cent. ideal efficiency is too high, and the accurate method of deducing this number depends on knowing the true adiabatic expansion line of the working fluid. Assuming the apparent specific heat to be the true specific heat, this line can be calculated from the tables on p. 235 by a method described in Appendix II.

The curve shown at fig. 106 has been so calculated for expansion from 1 volume to 10 volumes.

The curve is calculated on the assumption that a mass of gas which would occupy 5 volumes at a pressure of 14·7 lbs. per sq. in. at 100° C. is heated to 1830° C. at volume 1, and then expands adiabatically. The temperatures are marked on the diagrams.

Any other adiabatic between the temperatures on the line shown may be calculated from the diagram.

Calculating for the temperature conditions of the Clerk diagrams, shown at fig. 105, the ideal efficiency of the actual working fluid there used is found to be 39·5 per cent.

From this it appears that the air-standard efficiencies are too high. The actual properties obtained from these experiments only give 39·5 per cent. efficiency, while the air standard gives 49 per cent. As the engine balance-sheet shows 34·7 indicated efficiency $\frac{34 \cdot 7}{39 \cdot 5} = 0 \cdot 878$; that is, the actual enigne has converted 88 per cent. of the heat which it possibly could convert into indicated work. This shows that for given expansion the best engines have approached very closely to the theoretical realisation of their cycle. The complete suppression of all heat losses due to conduction, &c., on the explosion expansion strokes

could only increase the indicated power from 34·7 to 39·5—that is, improve it by about 13 per cent. This at once explains why so little difference has been found in the economy obtained from large engines as compared with small ones.

In order to check the new method by a test of an entirely different engine, experiments have been made in the author's laboratory upon a Stockport gas engine, $6\frac{1}{2}$ ins. diameter by 13 ins. stroke, giving about 5 HP. The engine is too small, and the heat loss is too large, to enable an accurate determination of specific heat to be made from that engine. The mixture used was slightly different, and the coal gas was

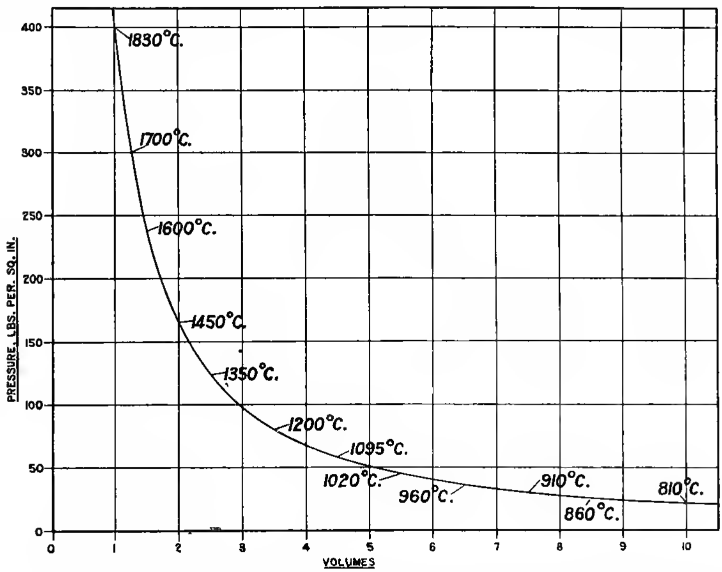

FIG. 106.—Adiabatic line for gas-engine working fluid with varying specific heats from table at p. 235 for a mass of working fluid which at volume 5 has a pressure 14·7 lbs. per sq. in. and temperature 100° C.

London gas instead of Ashton gas; but the specific heat values given in the tables on p. 235 are approximately applicable. Calculating from diagrams taken from this engine, and taking the mean of four cards, the following balance-sheet is obtained, calculated in foot-pounds and reduced to percentages:

	Ft.-lbs.	Per cent.
Indicated work	2,690	22·0
Heat lost on explosion and expansion . . .	3,880	31·7
Heat present in gases at exhaust . . .	5,350	43·7
Heat lost on compression	320	2·6
= 15·7 B.Th.U.	12,240	100·0

In this balance-sheet the heat loss is taken from the points AB on four

early-ignition cards. In fig. 107, A is a reproduction of one of the full-load cards, and B one of the early-ignition cards. The total heat present, as shown by the diagrams, is 15·7 B.Th.U.; the actual heat present, as determined by meter and calorimeter, is 14·5 B.Th.U.

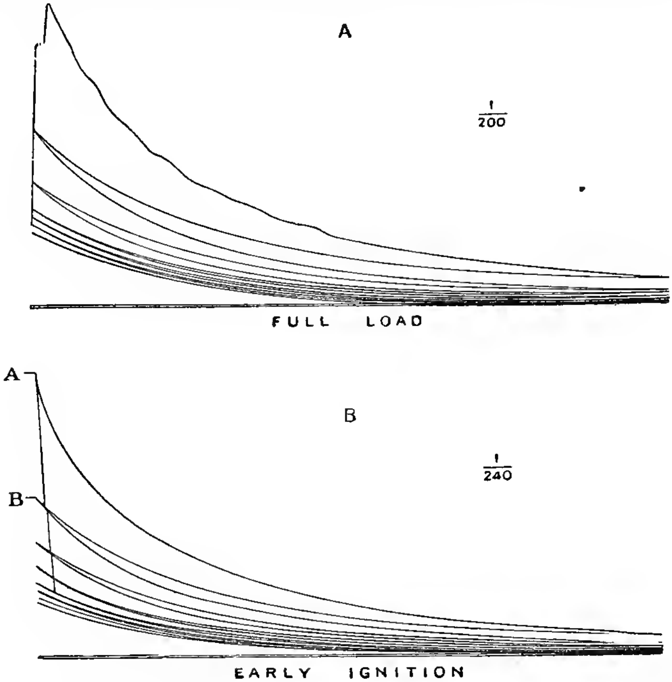

FIG. 107.—Clerk diagrams taken from a Stockport gas engine

An ordinary test made on the same Stockport engine in the author's laboratory under identical conditions gave the following balance-sheet:

	Ft.-lbs.	Per cent.
Indicated work	2,580	21·6
Heat lost to water jacket	5,920	49·6
Heat given up to exhaust calorimeter . . .	3,440	28·8
= 15·3 B.Th.U.	11,940	100·0

Comparing this with the balance-sheet obtained by the new method:

—	Ordinary trial	New diagram trial	
Heat flow during explosion and expansion and compression	49·6	31·7 2·6	34·3
Heat contained in gases at end of expansion . .	28·8		43·7
Indicated work	21·6		22·0

Here again the indicated work is practically the same in two trials, and the sum of the other two items is also very near. The distribution between the two varies as in the trials of the larger 'X' engine, but to a greater extent, a less proportion of heat going to the exhaust calorimeter in the smaller engine.

From this it appears to be proved that the apparent specific-heat numbers may be used for determining the approximate heat distribution in an engine, using the ordinary mixtures of gas and air for maximum economy.

Taking the apparent specific-heat values before given, efficiencies have been calculated corresponding to ratios of $\frac{1}{r}$ varying from $\frac{1}{2}$ to $\frac{1}{7}$.

IDEAL EFFICIENCIES

(With apparent specific heats given at page 235)

Suction temperature = 0° C., γ for compression line = 1·37

$\frac{1}{r}$	Maximum temperature 1600° C.	Maximum temperature 1000° C.	Air standard
$\frac{1}{2}$	0·195	0·200	0·246
$\frac{1}{3}$	0·286	0·293	0·360
$\frac{1}{4}$	0·354	0·356	0·430
$\frac{1}{5}$	0·384	0·394	0·480
$\frac{1}{7}$	0·439	0·443	0·550

This table also shows the efficiencies calculated from the air standard. From this it appears that roughly the air standard is about 20 per cent. too high, but within the range of practically applied expansions the air standard follows closely the ideal efficiencies given by the actual working fluid.

The author would point out that much knowledge is still required, and many determinations have to be made, before even the apparent specific-heat values can be sufficiently well known for all the various mixtures of inflammable gases used in gas and petroleum engines. When these numbers are sufficiently accurately established, each engine can be investigated separately, in order to determine the real efficiency of its working fluid and the margin remaining for improvement under given conditions. The inquiry is an exceedingly complicated one, which could hardly be undertaken with the time usually at the disposal of the manufacturing engineer.

A great deal of investigation must be performed before the numbers here given can be finally accepted as more than fair approximations.

Until further knowledge has been accumulated the air standard, as defined in the Committee's report, furnishes the easiest and best guide to enable the efficiencies of different engines to be predicted

or compared; indeed it furnishes a very delicate means of detecting error in design of either combustion space or valve proportions.

In using this standard, however, it must always be borne in mind, as the author pointed out as early as 1896, that the actual properties of the working-fluid differ from the ideal air which is accepted as a standard. In that year the author stated that, allowing for the properties of the actual working-fluid, an actual engine which gave an indicated efficiency of 0·277 could have given only 0·346 had heat loss been completely suppressed. That is, the actual engine gave 80 per cent. of its possible efficiency, while the air standard for the compression was 0·41, so that the relative efficiency on the air standard was 0·67.

For the purpose of comparison of different engines, this difference introduces no error; but for the purpose of investigation as to the actual limits of efficiency it has to be borne in mind that the working fluid can only give lower absolute efficiencies than those determined by the standard. But if γ be taken as 1·285 for the explosion expansion-line, and 1·37 for the compression line, the change of apparent specific heat between the temperatures of 1700° and 1000° C. commonly used in practice is too small to introduce serious error.

Much light has been thrown upon the question of thermal efficiency—both indicated and brake—by the Institution of Civil Engineers' Committee's experiments, and the author hopes that his own later contribution to the problem may also prove to have thrown further light on some of the difficult matters remaining, but undoubtedly both investigations suffered to some extent because of indicator imperfections. As his new method requires extreme accuracy of indicator, the author has designed a new optical indicator with which he is continuing his investigations on the same three engines, 'L,' 'R,' and 'X,' of the Institution Committee; the results, however, are not yet ready for publication.

Professor Hopkinson has also recognised the necessity for accurate indicator work, and he has produced a new optical indicator of very simple construction, with which he has performed two useful and important investigations now to be described.

Hopkinson's Experiments on Mechanical Efficiency with a Crossley Engine and Optical Indicator[1]

The dimensions and other particulars of the engine used are as follows:

Cylinder	11½ ins. diameter × 21 ins. stroke
Speed	180 revolutions per minute
Compression space	407 cub. ins.
Compression ratio	6·37, that is $\frac{1}{r} = \frac{1}{6{\cdot}37}$
Compression pressure	175 lbs. per sq. in. absolute
Air standard efficiency	52·2 per cent.

[1] Institution of Mechanical Engineers, 1908.

The engine is intended to give a maximum output of 40 HP on the brake when exploding at every cycle at 180 revolutions per minute; the indicated horse-power is 0·495 times the mean pressure.

Hopkinson defines the terms 'indicated power' and 'mechanical loss' as follows :

'*Indicated power* is the area of the positive loop of the indicator diagram multiplied by the number of explosions per minute and by the appropriate constant for reducing to horse-power.

'*Mechanical loss* is the difference between indicated power and the brake power delivered at the circumference of the flywheels.

'It includes the negative loop of the working diagrams and also the negative work done when the engine takes no gas.'

These are good definitions, and the present author has long used the terms in the sense here defined.

In this engine Professor Hopkinson found that the diagrams obtained were remarkably uniform for many consecutive explosions when working at or near full load. Fig. 108 shows a series of one hundred consecutive explosions one above the other.

It is to be noted, however, that the compression in this engine is relatively high, and the ignition accordingly very rapid; such compressions necessitate the water-injection device adapted by Messrs. Crossley to this engine. Without the water-injection this engine would pre-ignite when running for any length of time with full load.

The experiments made with the Hopkinson optical indicator undoubtedly prove that accurate results are to be obtained by its use.

Hopkinson gives the following table showing the values obtained from three experiments with optical indicator and brake :

INDICATED AND BRAKE HP OF CROSSLEY $11\frac{1}{2} \times 21$ INS. ENGINE. (*Hopkinson*)

Speed . . . . 180 revs. per minute.

Water jacket; exit temperature	$\frac{\text{Explosions}}{\text{Cycles}}$	M.E.P. from diagram	Gas per suction stroke	IHP	BHP
			Cub. ft.		
150° F. (65° C.)	0·804	100·3	0·1196	39·7	34·0
—	0·82	99·4	0·1182	40·2	34·6
160° F. (71° C.)	0·825	99·0	0·1164	40·2	34·9

The mean of the three observations of brake horse-power is 34·5, and the mean indicated power 40, which gives a mechanical efficiency of $\frac{34·5 \times 100}{40} = 86·2$ per cent.

The mechanical loss of the engine with this load, and running at 180 revolutions, is thus 5·5 HP.

Hopkinson found that the mechanical loss varied considerably with

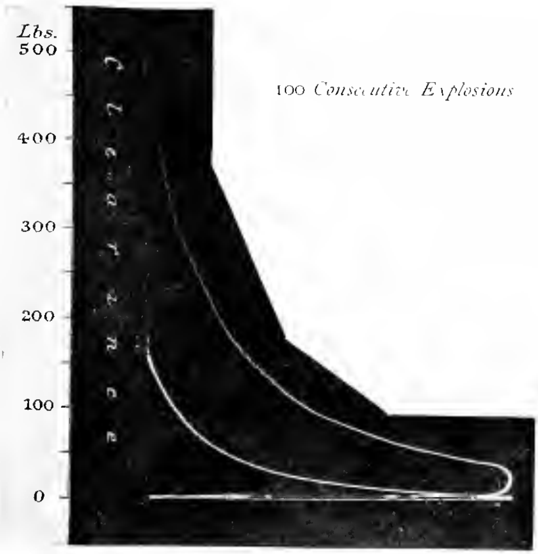

FIG. 108.—Diagrams taken from 40 HP Crossley Engine by Hopkinson's optical indicator.

the temperature of the water jacket; when the jacket was cold the loss was greater than when the jacket was hot. In order to investigate this point independently of either indicator or brake, he arranged a dynamo to motor round the engine at its normal speed of 180 revolutions per minute, and measured the power taken to drive the dynamo electrically; after deducting the dynamo losses this gave the power required to motor round the engine plus the loss in the driving belt, estimated at about 0·5 HP.

He varied the temperature of the water jacket from 21° C. to 82° C., and made experiments with normal lubrication, excess of lubrication, and water injection, in addition to ordinary lubrication. During these experiments the exhaust-valve cover was removed, so that the cylinder was opened to the air and there was no loss from air resistances or from compression. The results obtained were as follows:

FRICTIONAL LOSSES OF CROSSLEY $11\frac{1}{2} \times 21$ INS. ENGINE WITHOUT PUMPING LOSSES DETERMINED BY ELECTRICAL DRIVING. (*Hopkinson*)

Engine motored at 180 revs. per minute

	Power absorbed HP
Engine hot (about 180° F., 82° C.), normal lubrication	4·0
Engine cold (70° F., 21° C.), normal lubrication	6·5
Engine cold (70° F.), excess of oil	4·7
Engine cold (70° F.), water injected	2·7

From this it will be seen that the purely frictional losses of the engine vary considerably with the temperature; for equal and ordinary lubrication a temperature of 21° C. causes the absorption of 6·5 HP to motor round the engine at 180 revolutions, while only 4 HP is absorbed to do the same thing when the temperature is 82° C. Excess of oil, however, even at the low temperature, brings the friction down to 4·7 HP. It is remarkable to note the great drop of friction by the injection of water into the cylinder—it drops at once to 2·7 HP. A separate determination of frictional loss was made with the piston and connecting rod removed; this included the main bearing friction, valve lifting, and driving belt losses.

It was found that:

Friction of main bearings, side shaft, valve gear, and driving belt, 180 revolutions per minute . . = 1·4 HP

If this 1·4 be deducted from the numbers just given, it is thus found that piston and crank-pin friction varied in these experiments from 1·3 to 5·1 HP, but the normal value of the piston friction with the jacket at 82° C. was

$$4 - 1{\cdot}4 = 2{\cdot}6 \text{ HP}$$

For normal working of this engine at 41 IHP, which is nearly full load, at the best jacket temperature of 82° C., Hopkinson determines the

mechanical efficiency as 87·8 per cent., of which the particulars and division are as follows :

MECHANICAL EFFICIENCY OF $11\frac{1}{2} \times 21$ INS. ENGINE (CROSSLEY) AND DIVISION OF MECHANICAL LOSS AT NEARLY FULL LOAD AND 180 REVOLUTIONS PER MINUTE. (*Hopkinson*)

Indicated HP = 41 ; Brake HP = 36 ; Mechanical Efficiency, 87·8 per cent.

Suction (pumping loss)	1·4 HP =	3·4 per cent. of IHP
Piston friction	2·5 HP =	6·1 ,, ,,
Other friction valve lifting . . .	1·1 HP =	2·7 ,, ,,
Total mechanical loss . . .	5 HP =	12·2 ,, ,,

The apportionment between piston and other friction is somewhat uncertain, because there is no load on the piston when the engine is motored round and the compression and driving load may increase the friction to some extent ; but the total friction and air losses determined by motoring round agree satisfactorily with the mechanical efficiency determined under the different conditions by the indicator and brake. Thus a series of experiments were made on different dates with normal lubrication but varying jacket temperatures, the proportion of working cycles to total cycles varying from 0·80 to 0·83 as before, and the speed being 180 revolutions per minute. The results were as follows :

MECHANICAL LOSS DETERMINED BY OPTICAL INDICATOR AND BRAKE UNDER VARYING CONDITIONS. (*Hopkinson*)

Date	Jacket temperature	Mechanical loss
Aug. 16, 1906	150°–160° F. (65°–71° C.)	6·0 HP
Aug. 22, 1906	185° F. (85° C.)	5·0 HP
Aug. 22, 1906	185° F. (85° C.)	4·9 HP
Jan. 31, 1907	69° F. (20° C.)	7·1 HP
Jan. 31, 1907	150°–160° F. (65°–71° C.)	5·5 HP
Jan. 31, 1907	203° F. (95° C.)	4·5 HP

Here we find a very satisfactory correspondence between the mechanical loss determined by motoring round and by indicator and brake : in both cases it amounts to 5 HP with a jacket temperature of 82°–85° C.

It may be taken, then, that under normal conditions and nearly full load the mechanical efficiency of this engine is about 88 per cent.

Professor Hopkinson's conclusion from his own experiments is that the mechanical efficiency of this engine varies from 86 to 90 per cent., according to jacket temperature.

Following the Institution of Civil Engineers' Committee experiments, already described, Hopkinson has determined the mechanical efficiency by indicating at full speed without load and determining brake power

also at full speed. The temperature of the jacket was 71° C. during the brake test, and was kept up near to that temperature for the no-load indications. He found as follows :

	HP
Brake power	34·5
Indicated power, no load	7·35
Total IHP	41·85

Thus the indicated power obtained by the method recommended by the Committee would be 41·85 IHP. But by his indicator IHP was only 40 HP, so that the Committee's method gave 1·85 HP too much, and, had the engine been running exploding every cycle this would have amounted to an excess of 2·3 HP.

This he accounts for by the well-known fact that the resistance to the discharge of air drawn in during an idle cycle is greater than that due to hot exhaust gases; further, the expansion line of the compressed air is always a little lower than the compression line. He shows that in this engine the mean fluid resistances, suction and exhaust discharge loop, during firing, is 2·9 lbs. per sq. in., while during idle strokes it is 5 lbs. per sq. in., or 1·4 HP resistance against 2·5 HP. The compression and expansion loss he estimates at 1·5 to 2·5 lbs. per sq. in., adding a further 1 HP to the negative work, so that the total air resistances running without load should be about 3·5 HP; $3·5-1·4 = 2·1$ HP more than at nearly full load. This accounts nearly for the 2·3 HP before referred to.

Hopkinson made two experiments by electrical driving to throw light on this point. The engine was motored at full speed in both cases : with exhaust valve cover removed, and with the cover closed down. In the latter case, of course, the full air resistances would be experienced.

The following results were obtained :

	Aug. 24 HP	Aug. 25 HP
Engine closed	7·72	7·1
Engine opened	4·14	3·77
Difference	3·58	3·33

The mean air resistance obtained in this way is 3·45 HP, which compares satisfactorily with the 3·5 HP estimated by the indicator.

Because of these results Professor Hopkinson criticises the Committee's method, and points out that while the true mechanical efficiency was 86·2 per cent., as determined by indicated and brake HP, by the Committee's method in the particular case it would have been

$$\frac{34·5 \times 100}{41·85} = 82·5 \text{ per cent};$$

that is nearly 4 per cent. too low, while, if ignitions had been con-

secutive, it would have been 5 per cent. too low. In this particular Crossley engine it is true that there is a very considerable difference between the pumping losses under load and running light; yet this is not true of the particular engines criticised—that is, the 'L,' 'R,' and 'X' engines of the Institution of Civil Engineers' report. In these engines the difference is so small that it may be safely neglected. The total resistance of the 'X' engine, as determined by the indicator when running without load at 176 revolutions per minute, is only 6·9 HP. Remembering that the cylinder is 14 ins. diameter and stroke 22 ins., this compares favourably with the 6 HP, which is the mechanical loss of the $11\frac{1}{2}$ ins. by 21 ins. Crossley engine.

The 'X' engine gives air resistance, when running without load, of about 2 lbs. per sq. in., and, when running with full load, about $1\frac{1}{2}$ lb. per sq. in., both tested by a light spring. The difference only amounts to $\frac{1}{2}$ lb. per sq. in. mean pressure.

The fact that the mechanical efficiency of this 'X' engine, tested in the Committee's manner, is 0·89 per cent. proves that little can be lost in pumping.

It is the author's view that the Committee's method should give rather too high values for mechanical efficiencies because of increased friction of the engine under heavy loads, and accordingly he prefers to accept the mean of the two Committee methods given at p. 251, and take the mechanical efficiency of the three engines as

L	R	X
84 per cent.	85 per cent.	86 per cent.

under ordinary lubrication and jacket temperature of 150° and 160° F. (65° to 71° C.).

These values agree excellently with Professor Hopkinson's values for the 40 HP Crossley engine of 86 per cent. to 90 per cent. mechanical efficiency.

Hopkinson's Experiments on the Mechanical Efficiency of a Petrol Motor

Professor Hopkinson has also determined the mechanical efficiency of a four-cylinder petrol motor running up to 1,300 revolutions per minute.

The engine was made by the Daimler Co., Ltd., and has four cylinders, each 3·56 ins. diameter and 5·11 ins. stroke.

Professor Hopkinson describes his experiments as follows:

'The method consists in running one cylinder only of the engine and of indicating that cylinder, there being no load on the engine. The indicated power of the single cylinder is then equal to the mechanical friction of the engine plus the negative work shown on the

diagrams of all the four cylinders. When the engine is running fully loaded at the same speed, the loss by mechanical friction will be substantially unaltered, but the suction losses will not be quite the same owing to the same causes that operate in the case of the large gas engine. The suction loops in the three idle cylinders are different from those in the firing cylinder, and there is, moreover, the negative work done in the compression and expansion. The pumping losses, however, in the Daimler engine bear a very much smaller proportion

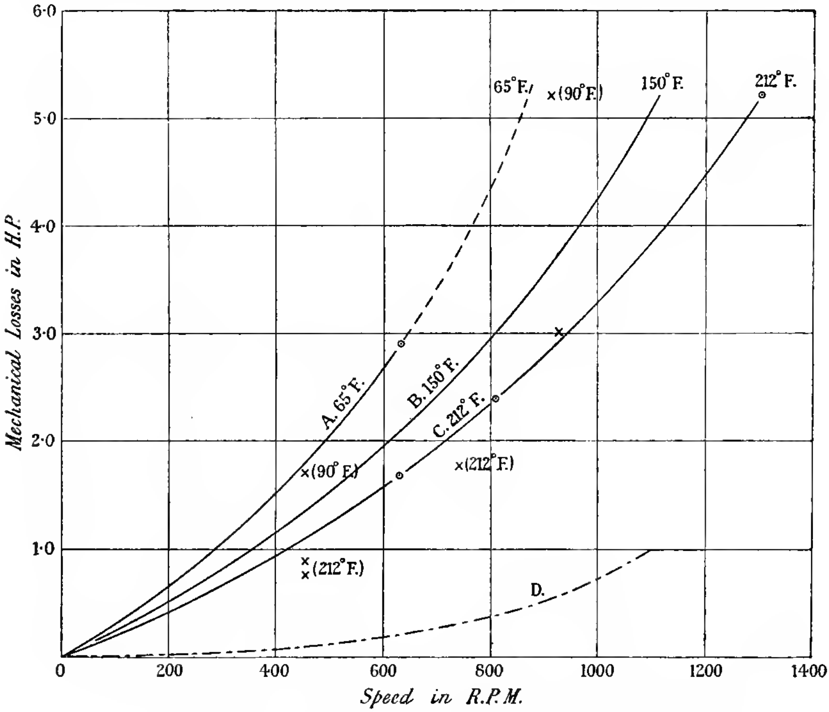

FIG. 109.—Mechanical loss of Daimler engine with varying temperatures of water jacket. (*Hopkinson*)

to the whole than in the large gas engine, owing to the relatively great size of the ports, so that no serious error is involved in neglecting the difference in these losses as between firing and not firing. Fig. 109 shows the relation between the power indicated by the single cylinder and the speed for three different temperatures. In curve A the outlet temperature of the jacket water was 65° F. (18° C.), in curve B about 150° F. (65° C.), and in curve C the water was just boiling. The dotted curve D shows the power absorbed in pumping, estimated from

light spring indicator diagrams; so that the difference between this curve and any one of the others gives an approximation to the loss by mechanical friction. These curves show that the frictional losses at a temperature of 65° F. are nearly double those when the jacket water is boiling. They also show that the losses increase very much more rapidly than in proportion to the speed, as is to be expected from the fact that they are due to fluid friction.

'Mr. Morse devised another method of getting the mechanical efficiency of a multi-cylinder motor, which, as it is very simple and appears to be quite accurate, is worth giving here. It consists in running the engine loaded with all the cylinders working. The load is put on by means of a Prony brake clamped to the flywheel, carrying a dead weight which is partially supported by a spring balance having an open scale. The spring balance reads the excess of the dead weight over the brake load in the usual manner, and small changes in the brake load may be very accurately read. In making a test one cylinder is stopped from firing by cutting off the current, and the pressure on the brake-blocks is reduced until the speed has come up to its old value. The reduction in brake load is then read off, and is approximately that corresponding to the indicated power of the cylinder which has been cut out. The four cylinders are treated in succession in this way, and by adding the results the indicated power of the engine is determined. Points obtained by this method with the corresponding jacket temperatures are shown by crosses (×) on fig. 109, and it will be seen that they agree very well with those obtained by indicating only one cylinder.

'The mechanical efficiency of this engine is remarkably high for its size. With boiling jacket water the mechanical efficiency is 90 per cent. at a speed of 400 revolutions per minute. It falls off to 75 per cent. at a speed of about 1,300 revolutions per minute. The thermal efficiency is also high, reaching over 26 per cent. under the most favourable conditions. The air-cycle efficiency corresponding to the compression ratio 3·85 is 41·5 per cent., so that the relative efficiency is 0·625.'

This first method applied to the petrol engine is really the Institution of Civil Engineers' Committee method applied in circumstances such as they thought correct—namely, where the pumping losses bear a small proportion to the total loss and vary but slightly with or without load.

Hopkinson's Experiment on Thermal Efficiency of a Crossley Engine with varying Strengths of Mixture

It has long been known that the maximum thermal efficiency was obtained with a weak mixture of inflammable gas or vapour and air.

This was first pointed out by the present writer in 1882, and the best mixture for non-compression engines was shown by him in 1886 to be about 1 of gas to 11 of air. Gas-engine builders have long been aware of this, and have adjusted their engines for any given compression to the point of maximum economy by reducing the gas proportion.

Burstall's experiments in 1901, as discussed by the author in 1904, showed that high-flame temperatures were associated with reduced economy, and the present writer made many experiments on dilute mixtures and low-temperature explosions to increase economy, which experiments he described in 1904 at Cambridge.

Professor Hopkinson has made interesting experiments on this subject with the same Crossley 40 HP engine on which he made the mechanical efficiency experiments just described. These experiments were made with the optical indicator. Speaking of this instrument Hopkinson states:

'It was found that this instrument gave results for the indicated power which were more consistent than those obtained with the pencil indicator; and these results agreed closely with those got by adding to the brake-power the power absorbed at light load under the same conditions of lubrication and jacket temperature, a proper deduction being made for the difference in the pumping work at full and light load. It is probable that the full load indicated power of this engine can be determined either by the new indicator, or from the brake-power and mechanical losses, correct to within 1 per cent. For the indicated power at light load reliance has to be placed on the indicator only, but the agreement between the two methods in the other case shows that it can be trusted to within 1 or 2 per cent.

'In the same paper a method of measuring gas consumption was described, which consisted in observing the fall of a small standard gas-holder during some fifty suctions of the engine. The general consistency of the results showed that the gas consumption could be determined in this way to within 1 part in 200.

'As it appeared probable that both measurements, that is, gas-supply and indicated power, were capable of considerable accuracy, it seemed worth while to make some tests on thermal efficiency. The points chosen for investigation were the effect of strength of mixture and of scavenging. The method used for measuring the gas was especially advantageous for this purpose, for it gave the actual volume of gas used in the series of forty or fifty explosions from which the indicator diagrams were taken, and the materials for a complete measurement could thus be obtained in a few minutes. Diagrams with three or four different gas consumptions could be got within an hour, during which time the calorific value of the gas would remain constant, so that the effect of changing the strength of mixture or of

scavenging by running without load could be very accurately determined.

'In each series the tests were all taken within two or three hours; the measurement of gas-supply being made as described in the last paragraph simultaneously with the photographing of the diagram.

'In measuring the indicated power, two diagrams, each covering about a dozen explosions, were photographed in every test. These photographs were integrated by planimeter direct from the negatives by two independent observers. In no case did the mean pressures so determined differ by more than 1½ lb. per sq. in. or 1·6 per cent., and the average difference was about 1 per cent. Thus the mean pressures given, which are the means of the two observations, may be taken as correct in every case to within 1 per cent.'

With mixtures varying from 8·5 per cent. to 11 per cent. gas of total cylinder contents, including exhaust gases, Hopkinson obtained the following indicated thermal efficiencies:

INDICATED THERMAL EFFICIENCY OF CROSSLEY 11½ × 21 INS. ENGINE WITH VARYING STRENGTH OF MIXTURE. (*Hopkinson's Experiments*)

Gas			M.E.P.	Indicated thermal efficiency	Load
No.	Cub. ft. per suction	Percentage of cylinder contents *			
				Per cent.	
1	0·1275	11·0	102·2	32·5	Full load
2	0·1147	10·0	98·4	34·7	" "
3	0·1005	8·65	90·2	36·5	" "
4	0·1275	9·5	108·4	34·5	Light load
5	0·1140	8·5	101·6	36·1	" "

* Calculated on the assumption that the full-load suction temperature is 100° C., and the light load 50° C.

This shows that, at full load and 11 per cent. of gas present in the cylinder contents, the indicated thermal efficiency was 32·5 per cent., and this increased to 36·5 per cent. when the mixture was reduced to 8·65 per cent. of gas. Eleven per cent. of gas is equivalent to 1 gas 8·08 volumes of air + other gases. 8·65 per cent. is equivalent to 1 gas 10·6 volumes of air + other gases.

Very similar results are given by the two light-load tests, but the effect of dilution is very well seen by comparing the volume of gas taken per suction. With the engine at full load, and taking in 0·1275 cub. ft. of gas per suction, the indicated thermal efficiency is 32·5 per cent., but with a light load, taking exactly the same gas per suction, 0·1275 cub. ft., the efficiency rises to 34·5 per cent. In the same way, comparing full load with 0·1147 cub. ft. of gas and light

load with 0·1140 cub. ft. of gas, we find a change from 34·7 per cent. to 36·1 per cent. The change is also readily seen in the M.E.P. obtained with similar charges.

No. 1 experiment gives 102·2 lbs. per sq. in M.E.P., and No. 4, with the same gas, gives 108·4; likewise No. 2 and No. 5 give respectively 98·4 and 101·6, although the gas charge is practically the same.

Fig. 110 gives the relation between gas consumption and mean effective pressure.

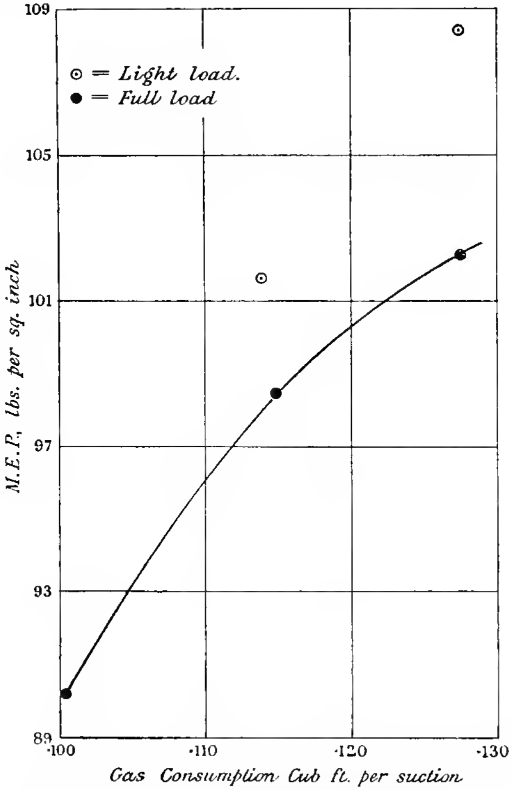

FIG. 110.—Curve showing relation between gas consumption and mean effective pressure, Crossley 40 HP engine. (*Hopkinson*)

With regard to these trials, Professor Hopkinson states:

'All the observations here shown were taken within two or three hours, and it may be assumed that the calorific value of the gas remained the same during that time.

'The calorific value was not taken at the time of the test, and the efficiencies are calculated on an assumed lower calorific value of

600 B.Th.U. per cubic foot at standard temperature and pressure—a value which accords with measurements made a day or two after the test. The relation between the efficiencies with different gas consumptions, which it was the main object of this experiment to determine, is unaffected by any error in the calorific value or in the calibration of the indicator, and is undoubtedly shown with great exactness in the above series of figures. That the absolute values are also fairly close was shown, however, by tests made at other times in which the calorific value was measured at the time of the test. In some trials the indicated power was calculated from the brake-power and the indicated power at no load, as in the Institution of Civil Engineers' trials, but with a proper deduction for the difference in pumping work as between light and full load. The following were the results of one such test:

Full Load

Gas used 0·1006 cub. ft. per suction at 52° F. and barometer 30·46 ins.
Calorific value (measured at time), 570 B.Th.U. per standard cub. ft.

$$\frac{\text{Gas}}{\text{Total Charge}} = 0{\cdot}0865.$$

$$\frac{\text{Explosions}}{\text{Cycles}} = 0{\cdot}896.$$

BHP = 34·4. Efficiency on BHP = 32·2 per cent.
Jacket temperature = 190° F.

Light Load (taken as soon as the brakes were off)

Gas, 0·1129. Mean pressure calculated from gas consumption, 100 lbs.

$$\frac{\text{Explosions}}{\text{Cycles}} = 0{\cdot}138. \quad \text{IHP} = 6{\cdot}9.$$

Extra pumping work, 0·896 × 2·3 = 2·0.

$$\therefore \left.\begin{array}{lcl} \text{Mechanical losses} & = & 4{\cdot}9\% \\ \text{Indicated horse-power} & = & 39{\cdot}3\% \\ \text{Mechanical efficiency} & = & 87{\cdot}5\% \\ \text{Thermal efficiency} & = & 36{\cdot}8\% \end{array}\right\} \text{at full load.}$$

The thermal efficiency at full load is plotted in terms of strength of mixture in fig. 111. The straight line which most nearly represents the mean efficiency is drawn through the points, about equal weight being given to diagrams and brake tests. A number of other tests are shown as well as those cited above.

The strength of mixture is calculated on the assumption that the suction temperature with full load is 100° C., and with light load (scavenged charges) 50° C. There is some uncertainty about these temperatures, and a corresponding uncertainty in the absolute value of the proportion of coal gas in the mixture. But as the total weight of charge is practically independent of the strength of mixture (if the engine be kept fully loaded and the jacket temperature constant)

the *relative* values of the proportions under full-load conditions are unaffected by this uncertainty. The only effect of an error in the suction temperature is to alter the horizontal scale of the diagram, fig. 111. The scavenged charges are dealt with later. The weakest mixture used in these tests contained about 8·65 per cent. of coal gas when in the engine, the proportion of air to gas drawn in being about $9\frac{1}{2}$ to 1. The diagram given was quite normal in appearance, the explosion line being nearly vertical. Fig. 112 is a reproduction of

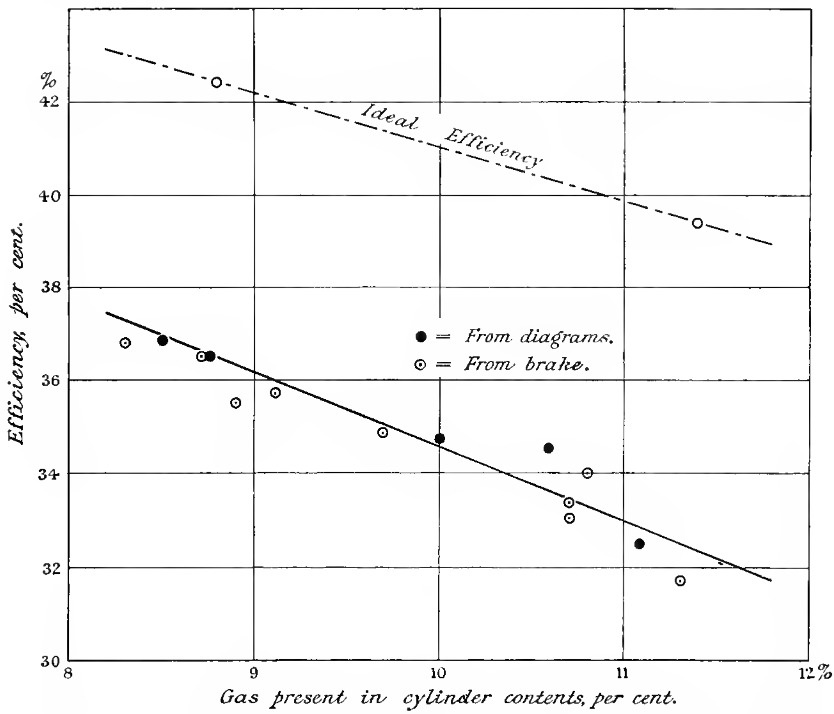

FIG. 111.—Curves showing indicated thermal efficiency at full load in terms of strength of mixture. (*Hopkinson*)

the diagrams corresponding to the extreme charges used. Weaker mixtures than this, however, would not ignite regularly. At the other end of the range the proportion of air to gas was about $7\frac{1}{2}$ to 1, the excess of air being about $1\frac{1}{2}$ times the volume of gas; slightly heavier charges than this could be used, but it is possible that the combustion would not be complete and the pressures in the engine would become dangerously high. The range of mixtures tested therefore covers all which could be practically used. Within that range the

efficiency diminishes steadily as the strength of mixture increases, the difference between the weakest and strongest charge amounting to $4\frac{1}{2}$ per cent. in efficiency, or 12 per cent. on the work done.

Hopkinson comes to the conclusion that this improvement of 4·5 per cent. in efficiency or 12 per cent. on the work done by using weak mixture instead of strong is due to (1) the change of specific heat of the charge by changing temperature; (2) the diminution of heat loss to

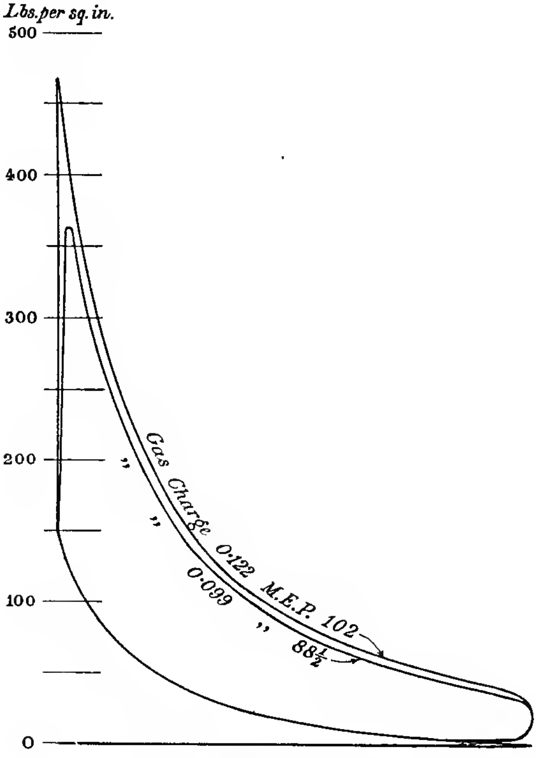

FIG. 112.—Optical indicator diagrams corresponding to extreme charges used. (*Hopkinson*)

the enclosing walls because of the reduced temperature of the working fluid when the gas charge is weak.

He discusses the division between change of specific heat and diminished heat loss by means of the curves shown in fig. 113, of which he says:

'The ideal efficiency of a gas engine, by which is meant the efficiency which would be attained if all heat losses to the walls were

suppressed and if combustion were complete and instantaneous at the in-centre, is easily calculated if the internal energy of the working fluid is known as a function of its temperature. It cannot be said that we yet possess this knowledge in any high degree of accuracy, but enough is known to enable an estimate to be formed of the effect of strength of mixture on efficiency. Fig. 113 shows the internal energy curves corresponding to the weakest and strongest mixtures used in these experiments. The ordinate of the curve is the quantity of heat in foot-pounds required to heat a standard cubic foot of the burnt products from 100° C. up to the temperature represented by the abscissa. These curves are calculated from the figures given by Langen for the specific heats of CO_2, H_2O and air, between 1500° and 1900° C., and from the results of Holborn and Austin, and Holborn and Henning, at lower

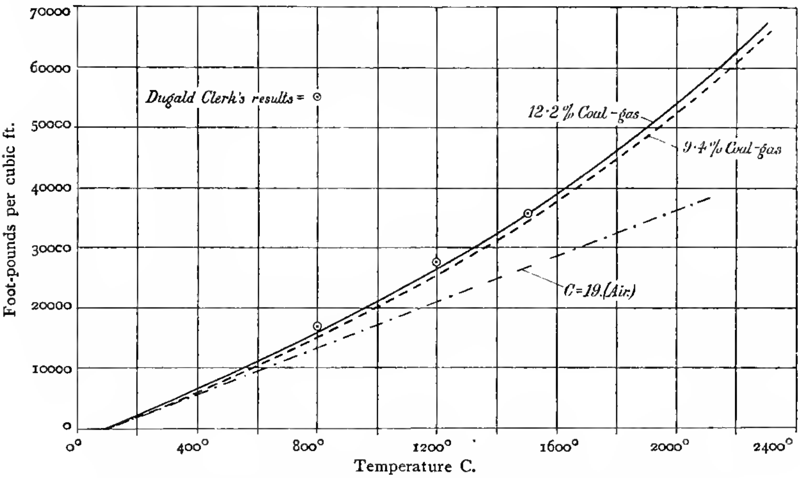

FIG. 113.—Internal energy curves for weakest and strongest mixtures used. (*Hopkinson*)

temperatures. The values given by Clerk for a mixture of intermediate composition are also shown. The ideal engine efficiencies for the two mixtures can be calculated from these curves by the method given by the author in the discussion on Mr. Dugald Clerk's paper before the Institution of Civil Engineers.[1] The ideal efficiencies corresponding to mixtures containing respectively 8·8 per cent. and 11·4 per cent. of coal gas, calculated by this method, are 42·4 and 39·4 per cent. respectively. For mixtures of other compositions the efficiency will follow a straight line law sufficiently nearly for present purposes, and this straight line is shown dotted in fig. 111. It is worth noting that the two straight lines on that figure, if produced, would cut the line corresponding to a

[1] "On the Limits of Thermal Efficiency in Internal Combustion Engines," Proceedings Inst. C.E., vol. clxix. page 157.

zero gas consumption, at 50·6 per cent. and 52·6 per cent. respectively. The air-cycle efficiency for this engine is 52·2 per cent. In other words, if it were possible to burn weaker mixtures—say, by using stratification—and if the actual and ideal efficiencies continued to bear a linear relation to the gas consumption, these efficiencies would tend to become equal to one another and to the air-cycle efficiency with a very small gas consumption. The ideal efficiency ought, of course, to approximate to the air-cycle efficiency when the charge is greatly reduced ; the close agreement in the other case is no doubt, to some extent, accidental.'

'Without laying too much stress on the absolute values [1] of the real and ideal efficiencies shown in fig. 111, it is apparent, from the ratios that they bear to one another, that, while much of the superiority of the weaker mixtures is to be ascribed to increase of specific heat, that cause is not sufficient to account for the whole of the effect. Comparing the actual with the ideal efficiency, it will be seen that for a mixture containing 8·5 per cent. of coal gas the ratio—usually called the efficiency ratio—is 0·87, but when the proportion of coal gas is increased to 11·0 per cent. it is only 0·83 ; the weaker mixtures, in addition to giving a higher ideal efficiency, come nearer in practice to realising that ideal. This is due to the fact that the percentage of heat lost to the walls during expansion is less with small gas charges than with large. The difference is sufficient to counterbalance an influence tending the other way, viz. the more rapid combustion of the stronger mixtures. This has been established by a series of experiments directed to that end.'

The author agrees with Professor Hopkinson as to the causes of the improvement with weak mixtures—it is undoubtedly due to diminished specific heat and diminished heat loss to the enclosing walls ; indeed, in the paper to which Professor Hopkinson refers, the author gives a table showing the change in ideal efficiency owing to the change in apparent specific heat, the maximum temperatures being taken as 1600° and 1000° C. respectively. At a compression ratio of $\frac{1}{5}$, the author's values show an improvement of over 2 per cent. due to this cause alone.

The author differs from Prof. Hopkinson in his calculation of the ideal efficiencies proper to the two mixtures: the ideal working fluid in the author's view should be taken as at the same maximum temperature as is attained by the actual explosion, not at the temperature which would have been attained had combustion been complete at the beginning of the stroke. In this way Hopkinson takes the ideal maximum temperature of the strong mixture to be 2210° C. and that of the weak mixture 1940° C., temperatures somewhat above those

[1] The *absolute* values of the efficiencies are affected by any errors in the calorific value of the gas or in the indicator calibration ; and may all be wrong in any experiment by as much as 1 per cent. But the relations between the efficiencies with different strengths of mixture will be unaffected by these errors, since they are based upon measurements with the same indicator and the same gas.

which would be given by the strong and weak mixtures used. The error introduced, however, for this particular purpose is too small to affect the accuracy of Hopkinson's conclusions.

In order to determine the proportion of the improved efficiency which is due to reduction of wall loss, Hopkinson has made experiments on the wall loss, which he discusses as follows :

'As pointed out by Mr. Clerk, the ordinary method of determining wall loss by the amount and rise of temperature of the cooling water does not give an accurate notion of the loss of heat occurring in the expansion stroke. Much of the heat in the cooling water passes into the walls after release, and should therefore in a proper heat balance be credited to exhaust. In a true heat balance the measured items must be the work done and the energy contained in the gases at the end of expansion, the heat loss during expansion being obtained by difference. Such a heat balance has been formed for the weakest and strongest mixtures used in these experiments.

'The energy at the end of expansion is in part thermal, and in part the chemical energy represented by unburnt gas. For the calculation of the first item the data required are :

'(1) Temperature of gas at end of expansion ;

'(2) Quantity of gas present ;

'(3) Its internal energy as a function of its temperature.

'The quantity of gas present is known from the suction temperature and suction pressure ; it may be taken as 1·06 standard cub. ft. per explosion in full-load running with a medium jacket. The internal energy is given by the curve, fig. 113. The temperature at the end of expansion can be inferred from the pressure of the gases at release. For measuring this the indicator was fitted with a large piston giving an open scale. A series of consecutive tests were made, the gas charges being alternately about 0·1 and 0·13 cub. ft. per explosion. In each test the gas charge was measured by gas-holder as described above, and the release pressure was determined simultaneously either by photographing the diagram or by reading it off in the telescope used with the indicator. The calorific value was also determined during the course of the experiments. The following table gives the mean of the results obtained in a series of such tests which show very good agreement :

——	A (weak mixture)	B (strong mixture)
Gas per explosion as measured by holder . . .	0·1007	0·1294
Gas used per explosion (std. cub. ft.)	0·095	0·122
Percentage of coal gas present in cylinder contents.	8·5	11·0
Pressure at release (lbs. per sq. in. absolute) . .	52	57
Pressure at end of expansion (lbs. per sq. in. absolute)	45	49·5
Temperature at end of expansion (absolute C.) .	1180°	1290°

From these experiments and his internal energy curves, fig. 113, Hopkinson calculated heat balance-sheets for the weak and strong mixtures A and B.

The following table shows these balance-sheets, to which has been added the balance-sheet of the 'X' engine of the Institution of Civil Engineers' trials as determined by the author's new method, see p. 269.

BALANCE-SHEET FOR CROSSLEY $11\frac{1}{2} \times 21$ INS. ENGINE WITH WEAK AND STRONG MIXTURE, TOGETHER WITH BALANCE-SHEET OF 'X' ENGINE BY CLERK'S METHOD

	Hopkinson's experiments		Clerk's experiment X engine
	Crossley A (weak mixture)	Crossley B (strong mixture)	
	Per cent.	Per cent.	Per cent.
Indicated work . . .	37	33	34·6
Heat in exhaust. . .	42	39	49·3
Heat loss in expansion .	21	28	16·1
	100	100	100·0

The exhaust gases were discharged into the exhaust calorimeter and instantly cooled down by water jets; the gases were collected and analysed. In five analyses at full load the percentage of fuel discharged unburned varied from 0·2 per cent. to 1·5 per cent., so that it may be taken that the results above shown are not materially affected by imperfect combustion.

The saving of heat due to the weaker mixture is thus seen to be 7 per cent., that is, 7 per cent. less of the total heat given to the engine passes through the walls during the explosion expansion stroke in the case of the weak mixture. Professor Hopkinson considers that this 7 per cent. would add 2 per cent. of the total heat to the work area, so that the 4 per cent. difference between the weak and strong mixture is due one-half to change in specific heat and one-half to saving in heat loss to the walls.

Hopkinson has compared the heat loss to the walls by the total energy method above described with the heat loss determined by the ordinary method of measuring the heat carried away by the jacket water for the two mixtures weak and strong, having charges respectively 0·1 and 0·13 cub. ft. of gas per explosion, and he finds that, measured thus, the heat loss is 27 per cent. and 33 per cent. Adding to this the radiation loss from the hot engine surface, he finds the total losses amount to 30 per cent. and 36 per cent. The values compare as follows :

—	A (weak mixture)	B (strong mixture)
	Per cent.	Per cent.
Heat loss on expansion	21	28
Heat loss by jacket and radiation . .	30	36
Difference	9	8

This shows that, determined by the ordinary jacket method, too much heat is trapped to the extent of 9 per cent. and 8 per cent. respectively of the total heat of the gas present.

This corresponds exactly with the difference shown by the Clerk method and the Institution of Civil Engineers' Committee method described at p. 270.

The two balance-sheets of the 'X' engine are given below:

	Committee's trials	Clerk's new diagram trials	Differences
	Per cent.	Per cent.	Per cent.
Heat flow during explosion and expansion	25·4	16·1	+ 9·3
Heat contained in gases at end of expansion	39·9	49·3	− 9·4
Indicated work	34 7	34·6	+ 0·1
	100·0	100·0	0·0

The 'X' engine trials thus show the difference between the jacket method and the Clerk new diagram method to be 9·3 per cent. of the total heat, corresponding to Hopkinson's experimental difference of 9 per cent.

It may thus be taken as fairly well established that the heat absorbed from the discharging exhaust gases by impinging on the water jacket round the exhaust valve, and the heat passing on the exhaust stroke, is in the case of weak mixture about 9 per cent. of the total heat.

Hopkinson states truly that no satisfactory method of determining radiation has yet been proposed, but he arrives at an approximation by determining the heat taken away by the water jacket when the exit temperature is 70° C., as compared with the engine running under exactly the same full-load conditions with the exit temperature at 40° C. He finds the difference to be between 2 per cent. and 3 per cent. of the total heat-supply; that is, between 2 per cent. and 3 per cent. of total heat-supply less finds its way with the jacket hot and jacket cold.

This, of course, assumes that the heat flow from the hot gases to the walls does not vary with the exit temperature of the water jacket. This the author fears is an assumption which cannot be truly made

even when the diagram shows but small differences between jacket hot and jacket cold. Hopkinson's figure of 3 per cent., however, cannot be far out, as the true radiation from the 'X' engine was determined by the Committee method as 2·4 per cent.

Hopkinson has made a number of interesting experiments with this

ANALYSIS OF EXHAUST GASES FROM CROSSLEY $11\frac{1}{2} \times 21$ INS. ENGINE UNDER DIFFERENT CONDITIONS OF RUNNING. ALSO GASES FROM BOYS' AND JUNKER CALORIMETERS

Test No.	Nature of Gas Analysed	1 Quantity of Gas used	2 Steam	3	4 CO_2	5	6 Percentage of unburnt coal-gas
		Litres	Mg.	Per cent.	Mg.	Per cent.	
1	Exhaust from Boys' Calorimeter $\frac{\text{air}}{\text{gas}} = 7·1$	1·53	0·8	0·3	−0·4	−0·1	0·1
2	Exhaust from Junker Calorimeter $\frac{\text{air}}{\text{gas}} = 9·6$	1·74	3·0	1·5	1·2	0·5	1·0
3	Engine Exhaust. Full load. Gas 0·1204 (11·25 per cent.)	1·5	4·0	1·8	2·8	1·2	1·5
4	Engine Exhaust. Full load. Gas 0·1228 (11·46 per cent.)	1·94	2·3	0·7	0·7	0·2	0·4
5	Engine Exhaust. Full load. Gas 0·1002 (9·37 per cent.)	1·63	1·4	0·8	2·5	1·3	1·1
6	Engine Exhaust. Full load. Gas 0·1320 (12·3 per cent.)	1·5	—	—	1·8	0·6	0·6
7	Engine Exhaust. Half load. Gas 0·1212 (11·3 per cent.)	1·5	2·8	2·7	6·3	5·7	4·2
8	Engine Exhaust. Half load. Gas 0·1199 (11·2 per cent.)	1·5	2·8	2·7	3·5	3·2	3·0
9	Engine Exhaust. Full load. Gas 0·13 (12·2 per cent.) .	1·0	1·3	0·7	0·3	0·1	0·4
10	Same samples as 9 . . .	1·72	1·5	0·5	−0·1	—	0·2
11	Engine Exhaust. Full load. Gas 0·1 (9·4 per cent.) .	2·0	1·2	0·5	1·5	0·7	0·6
12	Engine Exhaust. Half load. Gas 0·106 (10·0 per cent.)	1·56	3·3	4·2	5·6	6·8	5·5
13	Engine Exhaust. Half load. Gas 0·1285 (12·1 per cent.)	1·9	6·8	6·7	2·9	2·3	4·5

engine running about half load with the hit-and-miss governor, and so scavenging freely. He finds from exhaust gas analysis that at half load a considerable quantity of the gas supplied escapes unburned. He made four analyses of the exhaust when the engine was missing every other stroke, and found 4·2 per cent., 3·2 per cent., 5·4 per cent. and 4·5 per cent. of the total gas unconsumed, the average of the four experiments being 4·5 per cent. This is a most interesting and

unlooked-for effect of scavenging, and the deficiency is fully proved because of five trials with half load, in which all quantities were determined necessary for a balance-sheet, he found an average deficit of 10 per cent. of the total heat on the higher calorific value.

Hopkinson also determined the percentage of unburned gas leaving calorimeters of the Boys and Junker type.

The table on p. 294 gives the result of these important experiments.

Many other interesting and valuable facts are given in this important paper,[1] to which the author would refer the reader.

Professor Hopkinson is to be congratulated on adding very materially to our exact knowledge of the subject by his careful and ingenious study of a 40 HP Crossley engine.

PROFESSOR BURSTALL'S EXPERIMENTS ON A PREMIER GAS ENGINE OF 16 INS. DIAMETER CYLINDER AND 24 INS. STROKE, USING MOND GAS AND VARIOUS COMPRESSION RATIOS.[2]

The Gas Engine Research Committee of the Institution of Mechanical Engineers was originally formed in the year 1897, and Professor Burstall, of Birmingham University, was appointed reporter, to conduct the experiments and report to the Committee.

The present constitution of the Committee is as follows: Chairman, Sir Alex. B. W. Kennedy. Members: Messrs. Fielding, Humphrey, Burstall, Dugald Clerk, and Captain Sankey. Reporter: Professor Burstall.

Two reports have been already presented—in 1898 and 1901—but these were concerned with a small engine of 6 ins. diameter and 12 ins. stroke.

Professor Burstall's present experiments were made on a specially modified Premier engine operated with Mond gas at compression ratios varying from $\frac{1}{4\cdot36}$ to $\frac{1}{8\cdot07}$. The smaller compression space thus gave compression of over 200 lbs. per sq. in.

Professor Burstall thus describes the object of the tests:

'The tests were undertaken to determine in the first place the thermal efficiencies based on the indicated horse-power at various compressions, having regard to the richness of mixture, and in the second place to formulate if possible the law connecting efficiency and compression. Thus at each compression it was proposed to run a series of trials with different mixtures, which was done by using a number of different mixing valves in which the ratio of the air and gas ports varied. Had the composition of the gas throughout the tests been uniform this would have been a simple matter, but as the producer plant was in general worked at a fairly light load, it was

[1] Institution of Mechanical Engineers, 1908. [2] Institution of Mechanical Engineers, 1908.

impossible to ensure beforehand that the composition of the gas should be exactly what was required for the particular valve employed. The calorific value of the gas aimed at throughout the tests was 160 B.Th.U. per cubic foot (lower value).

'In the first instance a number of preliminary tests were run with various ratios of compression in order to obtain some idea of the engine generally and of the degree of accuracy with which experiments could

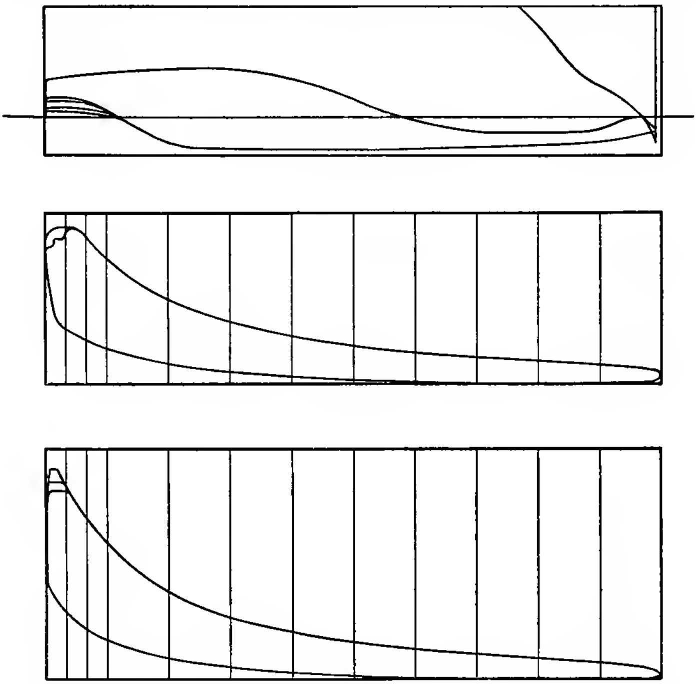

FIG. 114.—Indicator diagrams, Institution of Mechanical Engineers' experiments. (*Burstall*)

be made. As was only to be expected in a novel engine a considerable number of changes had to be made, particularly in the governing mechanism. New air and gas valves, with an entirely new arrangement of cams for driving them, had to be constructed, so that the engine as used for the trials differed very materially from the ordinary Premier engine, to such an extent, indeed, that it should be discussed entirely on its own merits.

'During the preliminary trials several types of indicators were used, including the Wayne rotary which was employed on the former trials, when using coal gas. This, however, was found to be so sensitive to the small quantities of dirt present with the producer-gas that its

use had to be abandoned, and the Crosby indicator was used instead. All the diagrams were taken by Mr. J. F. Gill, the reporter's assistant, the greatest care being exercised, in view of the fact that the indicated power was the most important measurement to be made. The barrel of the indicator was driven by a steel wire connecting the reduction gear on the piston to a bell crank about a foot above the indicator, from which it was driven by a very short string, thus getting rid of any errors due to the string itself. The diagrams were taken on smooth surface writing paper with a 6 H drawing pencil sharpened to a fine point, and before taking each diagram the indicator piston was lubricated. Three diagrams are reproduced on fig. 114. The spring employed was calibrated under steam pressure against a standard gauge, which itself had been calibrated against a dead-weight tester. The whole of the diagrams for a given test were measured up by the method of ordinates, and the mean of these ordinates plotted on squared paper. In this way it was found that a regular curve passed through all the points, and that any oscillations in the expansion lines caused by inertia cancelled out. The consistency of the diagrams is shown by the two trials F 2 and F 5, in which the indicated power is very nearly the same, while both the calorific value and quantities of gas widely differ, and, at the same time, the thermal efficiency of the engine has the same value. The power required to drive the engine itself was estimated from electrical data to be about 22 horse-power at full load, and from similar data the power empty to be about 20·8 horse-power. In all cases the indicated horse-power is calculated from the actual mean effective pressure on the piston. Reproductions of a number of diagrams were given in the paper. Losses due to back pressure and suction, and also the work required in the scavenger cylinder, are given in the tables.

'The compression of the engine was varied by inserting packing pieces at the big end of the connecting-rod, so that the compression space was always cylindrical in form, thus removing one of the defects of the former set of trials, in which the compression was varied by bolting a junk ring to the back of the piston. The variation in the mixture was obtained by using gas valves having different-sized ports for both air and gas, and also by throttling down the gas at the inlet. Although this latter method is very much more limited than could be desired, owing to the fact that it only alters the quantity of gas, it will be highly desirable in an engine of this kind to have the air for suction separate entirely from that for scavenging, so that both the air and the gas can be throttled during the progress of running the tests. This, however, was not foreseen at the time when the engine was constructed, and although the change could have been made, it would have involved a very considerable amount of alteration, which would have very much delayed the publication of the results.'

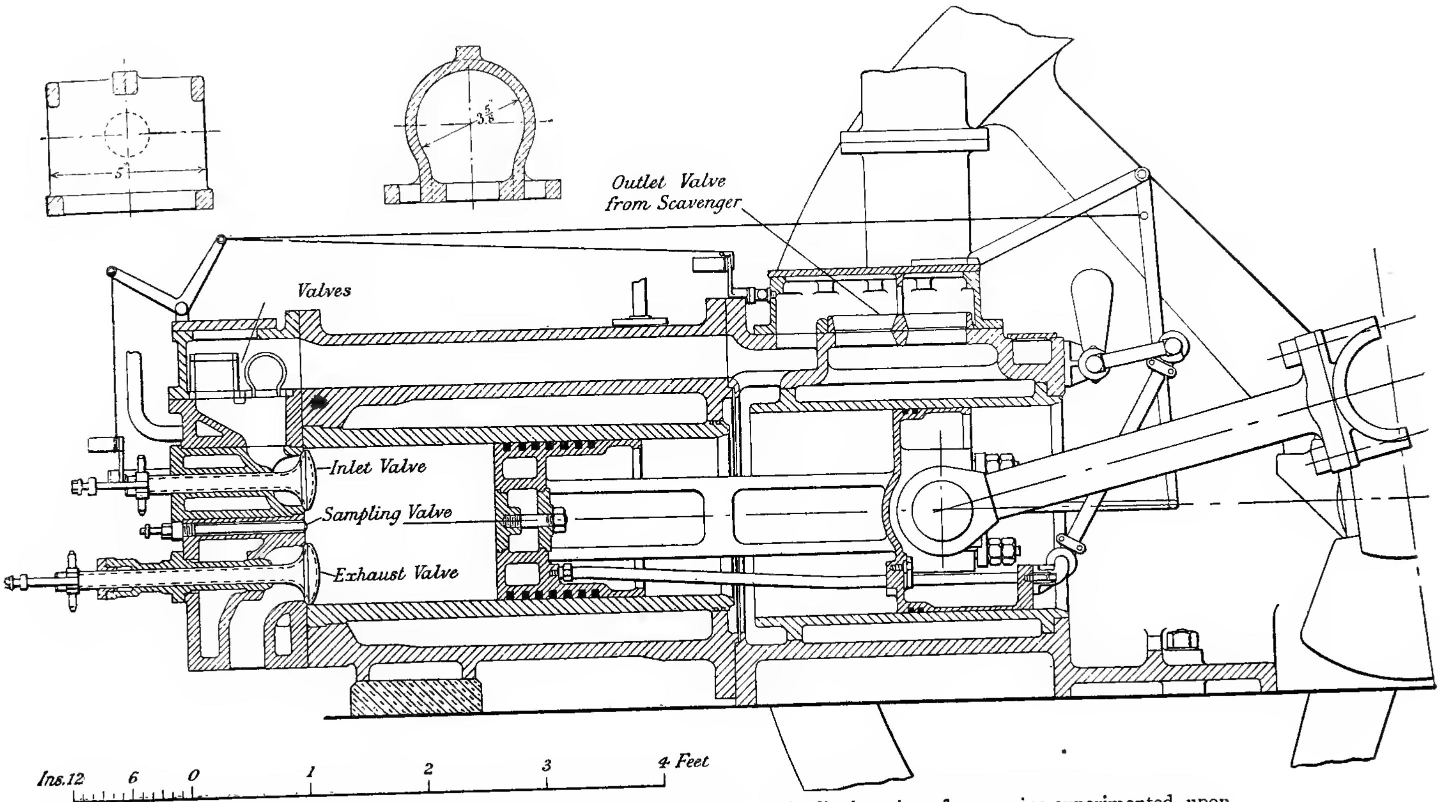

FIG. 115.—Institution of Mechanical Engineers' tests. Longitudinal section of gas engine experimented upon

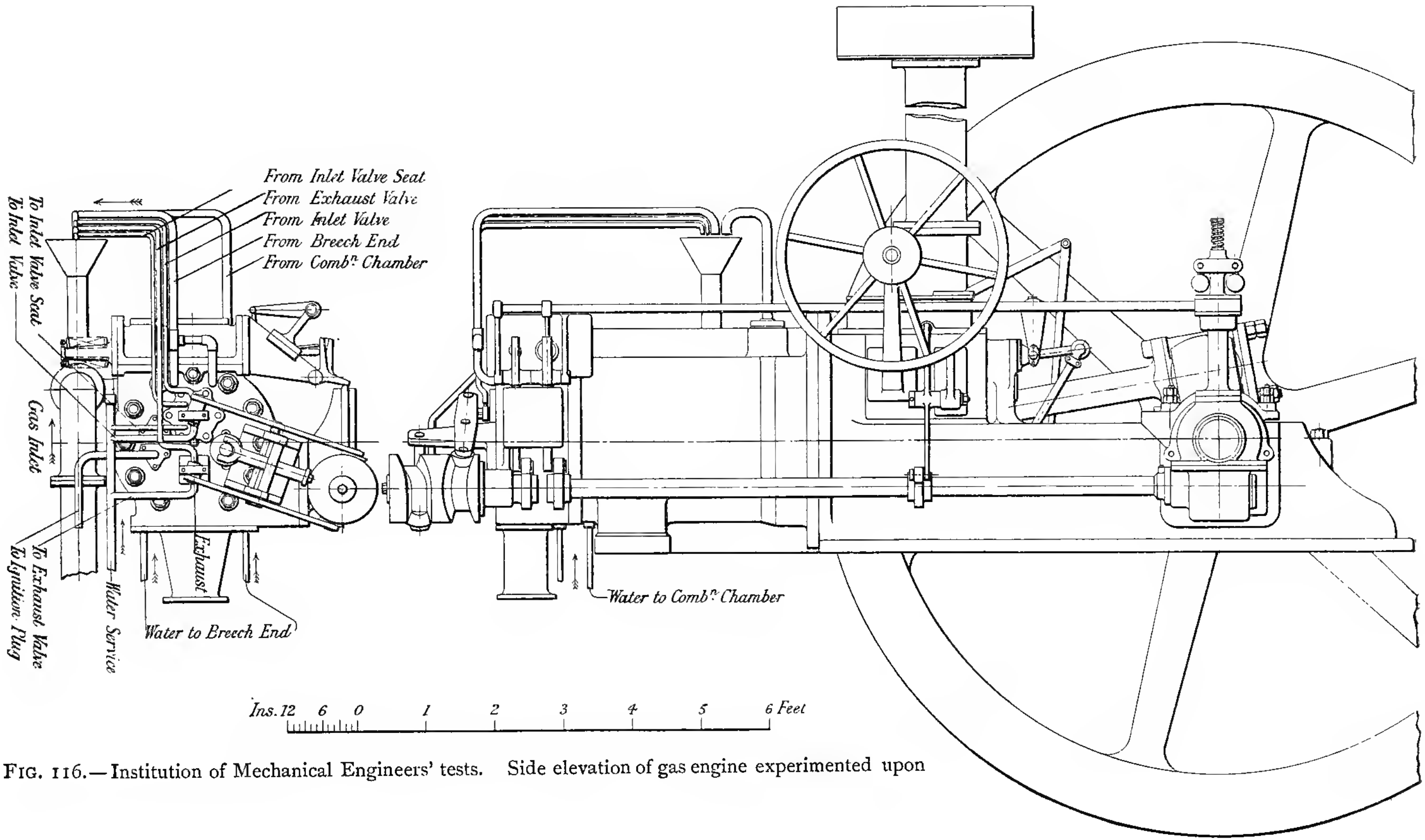

FIG. 116.—Institution of Mechanical Engineers' tests. Side elevation of gas engine experimented upon

The details of the engine experimented with are thus described :

'The engine chosen was that designated by the letter "O," capable of giving 150 horse-power at a speed of 170 revolutions per minute, the size of the cylinder being 20 ins. in diameter by 24 ins. stroke. In order to enable the engine to run at a compression pressure of 200 lbs. per sq. in., with charges which were estimated to give an initial pressure of 600 lbs. per sq. in., the diameter of the cylinder was reduced to 16 ins., and, at the same time, in place of using the standard admission valve on the top and exhaust at the bottom, an entirely new breech end was constructed, with the admission and exhaust valves horizontal, care being taken that the interior of the cylinder should have a perfectly flat end, like the cylinder of a steam engine. The cross sections and side elevation are illustrated in figs. 115 and 116, and a photograph is shown on fig. 117.

General Data

Particulars of engine	Ins.	Mm.
Diameter of piston	16	406·4
Diameter of differential piston	19	482·6
Stroke	24	609·6

Test	Clearance volume		Clearance surface		Ratio of compression
	Cub. in.	Litres	Sq. in.	Sq. cm.	
A	682	11·18	619	3992	8·07
B	726	11·90	632	4076	7·65
C	776	12·72	643	4147	7·22
D	833	13·65	657	4238	6·79
F	927	15·19	682	4399	6·20
J	1084	17·77	701	4521	5·45
Q	1436	23·52	807	5205	4·36

'The engine was so constructed that it could be worked on any one of the three known systems of governing—namely, (*a*) keeping the quality of mixture constant, and varying the amount ; (*b*) keeping the quantity of mixture constant, and varying the amount of gas ; (*c*) hit-and-miss, or cutting out of charges. The engine was originally arranged to work on system (*a*), but during the whole of the experiments it was arranged to work on system (*b*). As the tests are all at full power, the difference between the two is extremely small.

'The working of the engine is as follows : Starting with the suction stroke, the combined air-and-gas valve is opened to a predetermined point by a pivoted lever under the control of the governor and a positively driven pecker block, actuated by the half-speed shaft, the governor thus controlling the opening of the air-and-gas valve. The mixture, after passing through this valve, enters through the breech end into an annular casing, which contains the inlet valve, and

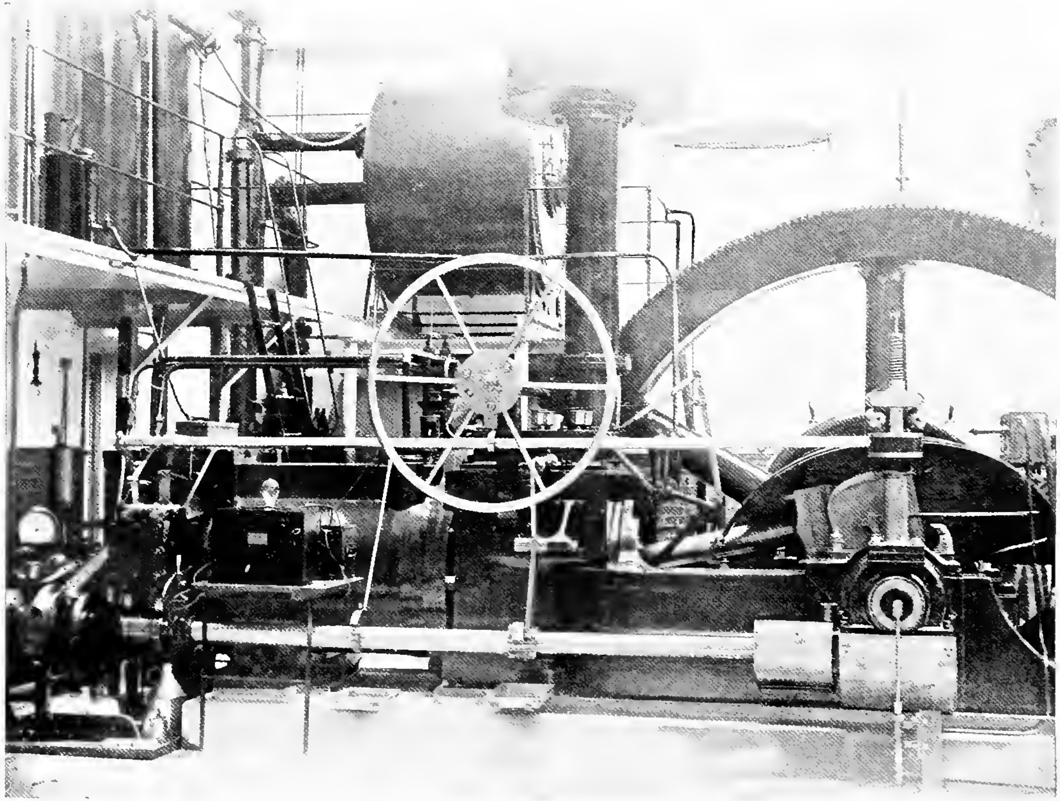

FIG. 117.—Institution of Mechanical Engineers' Tests. Photograph of engine experimented upon

then into the cylinder itself. After shutting the inlet valve the usual sequence of compression, explosion, expansion, and exhaust follows, but about half-way along the exhaust stroke a second valve, called the scavenger valve, lying alongside the mixing valve, is opened from the lay shaft, and allows a current of cold air from the differential piston to enter into the motor cylinder. This serves the double purpose of clearing out the exhaust products, and at the same time cooling the inner surfaces. During the idle stroke of the engine this scavenging charge is simply compressed and expanded in the passages leading up to the mixing and scavenger valves. In order to prevent, as far as possible, any possibility of pre-ignitions occurring through hot surfaces, every part of the engine exposed to the flame is water-jacketed, and, in order to estimate the amount of heat rejected through each of these surfaces, the water services are taken from separate measuring-tanks, and discharged without admixture with water from any other surface. The temperatures of discharge were in each case measured by thermometers placed in the outlet pipes. The number of separate services are as follows :

'(1) The barrel. This includes the water round the liner only.

'(2) The breech, which includes the water supplied only to the flat end of the cylinder end.

'(3) The piston

'(4) The inlet valve, inlet casing, and exhaust valve.

'As the engine was intended not only for experimental purposes but also to take its due share of the general work of the university, it was direct-coupled to a 55-kilowatt direct-current dynamo, built by the Westinghouse Company to supply current at from 110 to 125 volts.

'The engine works entirely with producer-gas made from bituminous fuel, supplied from a 500 horse-power producer of the Mond pattern, made by the Power Gas Corporation. After the gas has passed through the sawdust scrubbers, and entered the power house, it passes into a standard wet gas-meter, having a capacity of 8000 cub. ft. of gas per hour, made by Messrs. Braddocks, of Oldham. On issuing from the meter the gas passes through a Stott governor to the engine. Without the governor it was found that the suction of the engine produced violent oscillations of the water in the meter, which, besides affecting the accuracy of the readings, threw it out of order. Before commencing the tests the meter was calibrated against a standard meter by Messrs. Braddocks, and the water-level was kept constant throughout the whole of the experiments by running water continually through the meter.

'*Ignition.*—The engine was at first fitted with the ordinary make-and-break ignition, working from a 110-volts circuit. This worked quite satisfactorily until the compression pressure was raised to 140 lbs.

per sq. in., when premature ignitions resulted from the heating of the ignition points. This was more particularly the case when rich charges were used. As it would have been very difficult to use the magneto in the cramped position in which the ignition plug lay, high-tension ignition seemed to offer the most favourable opportunity of success, and of such the best seemed to be the Lodge ignition, which is so well known that a brief description will be sufficient. It consists of the usual induction coil and trembler, the primary circuit of which is made and broken by a contact on the lay shaft. This allows the timing of the spark to be readily adjusted while the engine is running, a matter of very considerable importance in an experimental engine. The secondary circuit is led to one of the coatings of a Leyden jar, the other coating of which is connected to the insulated central stalk of the ignition plug, one side of the high-tension circuit being, as usual, connected to earth. When the primary circuit is completed and the trembler in motion, a current passes through the secondary circuit, charging the Leyden jar and inducing another current on the opposite coating. When the potential is sufficiently high the coil discharges across a pair of outside points, and the Leyden jar itself discharges across the spark gap in the engine cylinder. According to Sir Oliver Lodge, the inventor, this "B" spark, being very rapidly oscillatory, is unaffected by water, dirt, or oil in the engine cylinder, and the reporter's experience of this system, extending over some two and a half years, is that it gives no trouble, provided the batteries which supply the primary current are always kept charged.

'Considerable trouble was experienced with the ignition plugs, particularly owing to the lack of mechanical strength in the earlier forms, as the sparking head was apt to shear away from the centre stalk. This was prevented by using a steel tube both screwed and brazed on to the head, which gave so little trouble that one plug was in use continuously for more than twelve months without cleaning or alterations.

'When, however, the compression was above 160 lbs. per sq. in., it was found that the central stalk of the ignition plug, which of necessity had to be insulated, became very hot, and the charge was apt to be fired from the hot metal. For the higher compression, therefore, the centre stalk was made hollow, and a stream of oil discharged direct on to the firing head in order to cool it, which entirely cured any premature ignition from this source. That this precaution was absolutely necessary was noticed during the running of the "A" tests, when once, the supply of oil having failed for a few minutes, distinct signs of premature ignition could be observed on the diagrams.

'*Results of the Tests*—It is not proposed to analyse at any length the results which are given in the tables, as it would take up not only a great amount of space, but might possibly lead to misapprehension on

account of having to employ constants, the values of which are not agreed upon by all authorities.

'In considering the thermal efficiencies, it will be noticed that for each compression there is a particular mean pressure which gives the highest economy for that compression. This pressure appears to range between 85 lbs. and 95 lbs. for all the compressions, with the tendency to increase as the compression goes up. Higher mean pressures than these caused the efficiency to fall off, as is shown in tests Q1, F2, F5. The rise in efficiency with the compression is very marked from the Q to the F trials, rising from a minimum of 28 per cent. up to a maximum of 39 per cent. After this point the efficiency increases comparatively slowly, reaching the maximum of 43 per cent. on the C trials, and then diminishing to 39 per cent. on the highest compressions of all. This result does not accord with the usual belief that economy increases with compression, when a suitable mixture is used. The cooling action of the walls, however, affects the result materially. Consider the contents of the cylinder at the end of compression. The gas is confined in a space 16 ins. in diameter, at the highest compressions about 3¼ ins. long, and at the lowest compressions about 6½ ins. long, the gas being entirely surrounded by water-cooled surfaces. This being the case, the leakage of heat during compression will be greater proportionately at the high than the low compressions, because the higher compression is accompanied by a higher density and by a temperature difference between the walls and the charge, and this more than compensates for the reduction of the area of surface exposed to the gases. Hence, after some definite compression is reached, further compression will result in a loss of economy and not a gain. For this particular engine the most economical compression pressure is apparently 175 lbs. per sq. in.; but, of course, the particular compression that will give the highest economy will vary according to the design of the clearance spaces, but it does not seem to be probable to get a design which will give better results than in the engine experimented upon. In order to obtain higher thermal efficiencies by the aid of higher compressions, it would be necessary to increase the stroke of the engine in proportion to its diameter. In the particular engine experimented upon the stroke is one and a half times the diameter. If the stroke were twice the diameter, it might be possible to employ a higher compression pressure. In this way the disc of hot gas might still be kept fairly thick, but, of course, such a method would mean slower speeds of rotation for a given piston speed, and thus it is quite probable than the lower speed of rotation might produce prejudicial effects, which would more than counterbalance the gain due to heat losses. Very high mean pressures, extending to some 114 lbs. per sq. in., were proved to be very decidedly uneconomical, the economy

falling from 39 per cent. to nearly 32 per cent. in the cases of F4 and F5. Such mean pressures were not used on the C, B, and A trials, because during the preliminary trials it was found that the economies were not good, and, moreover, that the maximum pressures produced —some 600 lbs.—were too high for safety. In the actual tests the maximum pressure that was allowed was 550 lbs., and this was only rarely reached. The heat rejected to the cooling water does not represent the whole of the heat lost to the walls, because the scavenger charge carries some portion of heat from the interior walls of the cylinder, and that heat is thrown into the exhaust. Hence, the values found for the heat rejected into the jacket water are lower than those which are generally obtained for non-scavenging engines.

'It will be noticed that in most cases the high thermal efficiencies coincide with the lowest percentages of heat lost in the cooling water; as, for instance, in the trials C1, D5, F4, while in the case of the A test at the very highest compressions of all, the quantities of heat rejected are greater than those in the B and C trials, thus corroborating, to some extent, the conclusion that it is possible to carry the compression too high in a particular engine.

'The thermal efficiencies throughout are computed by taking the heat values from the Junker calorimeter, which was kept running throughout the entire period of the running of the tests. The heat rejected to the different portions is as given in the tables, and it will be noticed that the barrel takes, roughly speaking, 50 per cent. more heat than the breech, that the piston takes about half the heat of the breech end, and the valves about three-quarters that of the breech. The explanation of the breech end taking more heat than the piston, although their surfaces exposed to the hot gases are the same, is the fact that the exhaust pipe passes through the watered breech end, and therefore abstracts heat during exhaust.

'Analysing the exhaust gases, it is possible to calculate the ratio of air to gas, and then, by assuming that the scavenger charge entirely removes all the products of combustion, to calculate the temperature at the end of the suction stroke. This has been done for the trials in which the mechanically operated sampling valve was used, and it gives the ratio of the air to gas varying from 1·6 to 3·6, and suction temperatures varying from 44° C. to 121° C. (111° F. to 250° F.). Further details are not given, because this method of deriving the value of ratio of air to gas from chemical analysis is subject to so many experimental errors as to render the figures so obtained not of much value, but probably for weak charges the suction temperatures will not be far removed from 50° C., rising to 100° C. for the rich charges. The maximum temperature calculated from these values was about 2000° C. (3632° F.), dropping to as low as 1200° C. (2192° F.) in all the economical runs of the tests given in the C trials.

'The whole of the experiments appear to point conclusively to the fact that the most economical mean pressure is very considerably below the maximum which can be obtained, and that the highest economies are obtained with a comparatively low maximum temperature. Both these results imply that the engine should not only be subjected to lower pressure, but to lower temperatures as well, and thus many of the difficulties which arise in large engines from rich charges might be avoided, and the maximum pressures kept down to quite reasonable limits.

'This, of course, only applies to the indicated power, and the conclusions as to the brake horse-power would be widely different. If, however, the engine is constructed to work only with these moderate pressures and temperatures, the whole of the working parts might be very much lightened, and thus a good mechanical efficiency obtained with the very moderate mean pressures.

'The question of the liability to premature ignitions, of gas containing larger or smaller percentages of hydrogen, was borne in mind throughout these experiments, but in every case of premature ignition which occurred—and many such cases occurred—with compressions higher than 160 lbs. per sq. in., it was traced to dirt or carbonised oil in the cylinder, or to some part having got overheated, and such premature ignitions took place equally with a weak as with a rich gas.

'The reporter is of opinion that, as far as premature ignition goes, the compression might be made a great deal higher than any which have been used during these experiments, but in view of the fact that the economy falls off after a certain point, there does not seem to be any useful object gained in going to any higher compression.'

The table on the next page gives the most important results; for other interesting tables the reader is referred to the original paper.

In this research Professor Burstall arrives at two main conclusions. First, that for any compression the highest economy is obtained when the mean effective pressure lies between 85 lbs. and 95 lbs. per sq. in. Second, that thermal efficiency increases from a compression ratio $\frac{1}{r}$ of from $\frac{1}{4\cdot36}$ to $\frac{1}{7\cdot22}$, and falls off at the higher compressions $\frac{1}{7\cdot65}$ and $\frac{1}{8\cdot07}$.

It is quite true that lower maximum and mean temperatures are the most economical, and under some circumstances that, of course, means that lower mean pressures are economical in the same engine as compared with high mean pressures. It by no means follows, however, that the limits always lie between 85 and 95 lbs. per sq. in., although no doubt they did so in the special circumstances of Burstall's

PRESSURES, POWER, AND THERMAL EFFICIENCY OF 16 × 24 INS. PREMIER ENGINE UNDER DIFFERENT COMPRESSION RATIOS. (*Burstall's Experiments*)

Test No.	Spark		Compression pressure		Mean explosion pressure		Mean effective pressure		Revolutions per minute	Indicated horse-power		Thermal efficiency average	Heat account		M.E.P. on suction	M.E.P. on scavenger reduced to area of piston
	Position of crank when contact was made, in degrees before dead centre	Distance of piston per cent. from end of stroke	Lbs. per sq. in.	Kgms. per sq. cm.	Lbs. per sq. in.	Kgms. per sq. cm.	Lbs. per sq. in.	Kgms. per sq. cm.		British	Metric		British Thermal Units per I.H.P.	Percentage to cooling water		
A 1	63	0·25	212	14·9	358	25·2	80	5·63	168	81·9	82·9	·394	6,431	23·9	2·81	0·6
A 2	54	0·24	222	15·6	384	27·0	86	6·01	168	87·6	88·8		6,485	20·5	2·72	
B 2	66	0·34	201	14·1	360	25·3	87	6·14	168	89·1	90·4	·398	6,360	18·0	1·90	0·8
B 3	68	0·36	191	13·4	330	23·2	82	5·8	168	84·1	85·2		6,439	16·4	1·64	
C 1	69	0·36	172	12·1	356	25·0	90	6·31	167	91·0	92·3	·415	5,930	17·1	—	0·7
C 2	48	0·19	175	12·3	324	22·8	92	6·49	171	96·1	97·4		6,384	21·5	1·67	
C 3	54	0·22	168	11·8	293	20·6	87	6·14	170	90·2	91·4		6,186	20·6	1·76	
C 4	46	0·18	175	12·3	343	24·1	99	6·97	170	103·0	104·2		6,069	23·2	2·32	
D 2	52	0·22	159	11·2	408	28·7	106	7·41	170	109·0	110·6	·404	6,295	23·2	2·15	0·7
D 3	54	0·33	166	11·7	388	27·3	110	7·70	170	113·0	114·7		6,398	23·1	4·45	
D 4	65	0·33	154	10·8	309	21·7	82	5·79	170	85·3	86·3		6,435	19·4	2·06	
D 5	60	0·28	158	11·1	334	23·5	93	6·52	173	97·8	99·1		6,110	12·2	2·1	
F 1	67	0·35	137	9·6	311	21·9	93	6·55	170	96·3	97·7	·359	6,485	16·7	2·1	0·6
F 2	38	0·12	129	9·1	430	30·2	114	8·02	167	116·0	118·0		7,839	25·8	2·2	
F 4	57	0·27	132	9·3	309	21·7	86	6·07	169	88·6	89·8		6,522	15·8	3·1	
F 5	34	0·10	129	9·1	354	24·9	113	7·93	169	116·0	118·0		7,757	22·4	3·0	
J 1	57	0·27	124	8·7	330	23·2	82	5·73	168	83·2	84·4	·345	7,315	30·9	2·68	0·5
J 2	36	0·11	132	9·3	414	29·1	106	7·47	167	108·0	110·0		7,520	27·3	2·75	
J 3	36	0·18	124	8·7	358	25·2	94	6·58	167	95·2	96·6		7,086	27·6	3·4	
Q 1	49	0·20	88	6·2	341	24·0	100	7·01	167	101·0	103·0	·300	9,012	27·8	2·6	0·5
Q 2	72	0·40	87	6·1	228	16·1	72	5·04	168	73·3	74·4		8,462	26·6	2·64	
Q 3	53	0·24	87	6·1	270	19·0	88	6·20	168	90·2	91·5		8,088	27·8	3·22	

experiments. As will be seen from the earlier discussion, the author is in agreement with this statement.

That thermal efficiency varies in the manner believed by Burstall with the varying compression ratio cannot, however, be accepted. Professor Burstall's experiments do not support his conclusions. The author stated this in his communicated remarks on the discussion as follows:

'Mr. Dugald Clerk wrote that Professor Burstall was to be congratulated upon the interesting experiments he had made on a gas engine of moderate power, driven by producer gas made from bituminous fuel. Such experimental work was, as he had explained, very difficult, because of the variation in the quality of the inflammable gas used during the experiments. The information given was therefore welcome, as supplying data of the working of gas engines with bituminous fuel gas, on which subject little exact experiment had been made.

'As a member of the Gas Engine Research Committee he had naturally taken much interest in the report, and in assisting to settle the experiments to be performed. These experiments were by no means so complete as was originally intended, inasmuch as the data given in the report was not sufficient to enable an accurate heat balance-sheet to be prepared for the engine. In the experiments made with three gas engines by the Committee of the Institution of Civil Engineers on the "Efficiency of Internal Combustion Engines," full sets of heat measurements were made for all heat quantities, gas entering the engine with its calorific value, heat carried away by the water of the water jacket, radiation from the external surface of the engine, and heat carried away by the exhaust as determined by an exhaust calorimeter. Brake tests were made also, as well as indicator tests, and by these experiments accurate deductions as to heat distribution were made possible. He was glad to say that it was determined by Professor Burstall and the Committee to repeat the "C" trials, those giving the highest efficiency, with brake tests, optical and mechanical indicators, exhaust calorimeters, and so forth, in order to obtain for these high efficiency values a sufficient body of data to make certain of accuracy of result. Notwithstanding the lack of determinations whch were originally intended, much useful information was contained in Professor Burstall's report.

'Professor Burstall, in the main, he was glad to say, corroborated the views which had long been held by those who had been engaged a large part of their lives in the practical work of gas-engine design. Speaking for himself, he was glad to have Professor Burstall's corroboration of a view which the writer had firmly held for many years on the subject of pockets or ports in the combustion space of a gas

engine cylinder. Professor Burstall's opinion was given in the first paragraph of the present report quoted by him from the second report to the Committee. He (Mr. Dugald Clerk) came to this conclusion long before 1895, in which year he discussed this matter at p. 307 of an edition of his book on the Gas Engine published in 1896. Summing up a discussion on some particulars of the Otto engine, he stated: "Indeed, it may be at once stated, that to gain the greatest advantage from high compressions the whole of the compressed explosive mixture should be contained in one space, that is, a space which is not divided into smaller separate spaces. Ports should be avoided if possible, and the flame should never be caused to flow through a narrow space into a wider one, as is done in this engine. The compression space should in fact be as nearly cubical or spherical as possible."

'In another part of the same book, a part published in 1886, p. 73, he stated: "The smaller the surface to which a given volume of working fluid is exposed the less heat will it lose in a given time. So far as loss of heat is concerned, then, the best type of engine is that which exposes a given volume of working fluid to the smallest surface in performing its cycle."

'On another point, he was glad to find himself in agreement with Professor Burstall—that is, as to limitations to the use of compression. Professor Burstall found that the most economical compression for this particular engine was 175 lbs. per sq. in., and that if higher compressions were to be used proportions must be altered.

'In 1895 he (Mr. Clerk) wrote in the same book, p. 385: "Such a compression as 210 lbs. per sq. in. above atmosphere would produce, with an explosion temperature of 1600° C., a maximum pressure of 675 lbs. per sq. in. above atmosphere, and this would involve an engine of nearly double the weight of working parts as compared with the engine tested by the author at Messrs. Crossley's, with but a small increase in power for the great increase in weight." "The author accordingly considers a compression of 200 lbs. per sq. in. as considerably above the limit likely to be useful in a simple gas engine; to render such compressions possible he considers that compound engines will require to be designed. The gas engine, in the author's opinion, is now rapidly nearing the limit of advantageous increased compression, so that no great further economy is to be expected there."

'This view was shared by many gas-engine designers, and it was gratifying to find Professor Burstall's experiments in some degree establish its correctness. On the point of greater economy possible from lower mean pressures, and therefore lower mean temperatures, he was also in agreement with Professor Burstall. Indeed, for many years it has been customary for gas-engine constructors to adjust their engines for the

point of maximum economy with a mixture giving much lower than the maximum possible mean pressures. That the weaker mixtures would give greater economies was, he believed, pointed out for the first time by the writer in a paper on Gaseous Explosions, read before the Institution of Civil Engineers in 1886. As a result of the examination of a large number of experiments, it was pointed out that a weak mixture containing $\frac{1}{12}$ of its volume of gas, that is, 1 of gas and 11 of air, gave the best result from the economical point of view, and that it had an advantage over that obtained from mixture having $\frac{1}{8}$ of its volume of gas, that is, 1 of gas and 7 of air. He then stated, p. 11 of the paper: "The total advantage of the weak mixture compared with the strong one is nearly as 100 to 72. The weak mixture is too long in attaining the maximum pressure, but in a gas-engine cylinder the slight mechanical disturbance caused by the gases flowing in at the port will cause the ignition to be sufficiently rapid."

'In the "James Forrest" lecture delivered by him at the Institution of Civil Engineers in 1904 he also called attention to the fact that low temperature, and therefore low-pressure mixtures, gave greater economy than high temperature and high-pressure mixtures.[1]

'So much for the points in which he found himself broadly in agreement with Professor Burstall. In detail, however, he differed somewhat from him. Professor Burstall stated in the present report (p. 10) that the rise in efficiency went on with increase of compression from 28 up to 39 per cent. thermal efficiency, but then it increased comparatively slowly, and reached a maximum of 43 per cent. on the "C" trials; and then it diminished to 39 per cent. at the highest compressions of all. In his view Professor Burstall's experiments did not establish the real existence of a 43 per cent. efficiency on the "C" trials. His experiments really agreed, if the mean line be taken, and the "C" and "D" trials be rejected, very closely with what would be expected from accepted thermodynamic considerations.

'To illustrate this, he had prepared the curves shown on fig. 118. In that figure the heavy dotted curve marked 1 was a fair curve through the values obtained from Professor Burstall's experiments, A, B, E, F, G. The light dotted line marked 2 showed the whole of Professor Burstall's experiments A, B, C, D, E, F, G, the points being the mean values, as given at Table I. of his report. It would be at once seen if experiments C and D were rejected, then the heavy dotted line, No. 1, very fairly passed between the points marking the other experiments. It appeared to him that the two experiments C and D were probably in error. It was impossible to draw a fair curve through Professor Burstall's experiments if these two results were included. The heavy black line marked 3 showed the values of the air standard

[1] Proc. Inst. C.E., vol. clviii., part iv., Table VI.

efficiencies for the different compression ratios multiplied by 0·71. This 0·71 was the number deduced by him from the Institution of Civil Engineers' experiments for use in his paper of last year on the limits of thermal efficiency in internal combustion engines. It expressed the relative efficiency of a gas engine of 14 ins. cylinder by

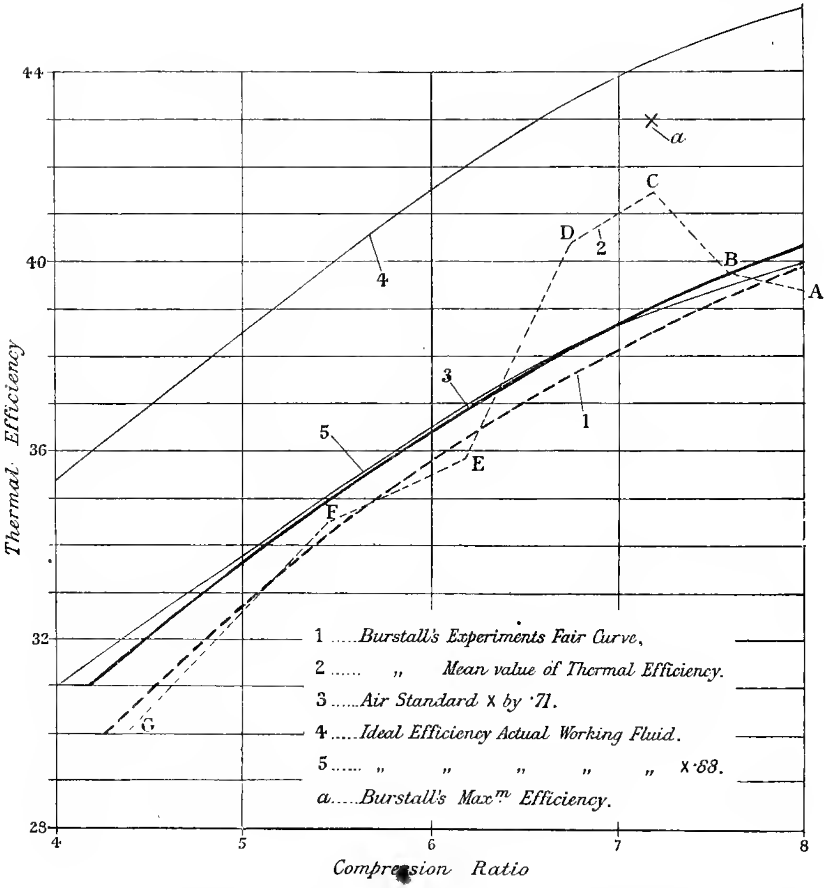

FIG. 118.—Curves showing thermal efficiency for varying values of $\frac{1}{r}$ (compression ratio), comparing Burstall's values with standards. (*Clerk*)

22 ins. stroke, as compared with the air standard. As Professor Burstall's engine was of 16 ins. cylinder diameter by 24 ins. stroke, it was to be expected that its efficiency ratio compared to the air standard would not depart far from this number 0·71. It would be observed that Professor Burstall's fair curve 1 very closely followed curve No. 3 the

air standard multiplied by 0·71. It differed very slightly from it at all points. For example, with a compression ratio of 5, Professor Burstall's fair curve 1 showed an actual efficiency of 32·7. The air standard multiplied by 0·71 was 33·6. At compression ratio of 6, Burstall's fair curve was 35·8; air standard No. 3, 36·3. Compression ratio 7, Burstall's fair curve, 38·1; air standard, 38·7. Compression ratio 8, Burstall's fair curve, 39·9; air standard, 40·3. The Burstall fair curve fell slightly below the air standard line No. 3.

'Taking another method of comparison: in a paper read before the Institution of Civil Engineers last year he (Mr. Clerk) deduced the actual properties of the working fluid by means of a series of experiments on apparent specific heat. The thin full line 4 showed the curve of efficiency for varying compression ratios, if the working fluid had the properties so deduced, and no loss of heat whatever occurred to the cylinder or enclosing walls.

'In the same paper, it was shown that the actual thermal efficiency attained by the engine investigated was found by multiplying the ideal efficiency of the actual working fluid by 0·88. The light full-line curve marked 5 gave the values of No. 4 multiplied by 0·88. This should give the values of thermal efficiency to be expected in an actual engine. Observe how closely this curve followed the air-efficiency curve No. 3, and how closely Professor Burstall's fair curve approximated to it throughout its whole range. The point *a* marked 43 per cent. was the highest point claimed by Professor Burstall, as determined by his experiments.

'From a consideration of these curves it seemed to the writer fairly obvious that some error had crept into Professor Burstall's experiments, series C D, and that the highest efficiency claimed by him, 43 per cent., was considerably higher than could be obtained with working fluid of the composition used by him, with anything like the heat losses noted in his experiments. If, for example, Professor Burstall had attained an actual indicated efficiency of 43 per cent., as shown at *a* upon the sheet, then he had come within 97 per cent. of the efficiency possible from his working fluid, when no heat loss occurred at all; that is, his heat loss with such an efficiency could not be more than about 8 per cent. of the total heat present. On looking at the heat losses given in experiment C, where the 43 per cent. was supposed to be attained, it was found to vary from 17·1 to 23 per cent., and these heat losses, it must be remembered, were heat losses which included neither heat carried away by scavenging air, nor radiation losses, so that they might very well represent less than the actual heat losses during expansion. The mean value of the heat lost to cooling water in the "C" experiments was 20·6 per cent. This cooling loss could not possibly have been sustained had the thermal efficiency been 43 per

cent. The number 20·6, however, agreed very well with the heat loss to be expected from Professor Burstall's fair curve 1. That curve showed that, at the point of compression corresponding to experiment C, the thermal efficiency shown by the experiments was about 87 per cent. of that possible with the working fluid used, and this corresponded very fairly to a heat loss value of 21 per cent., as heat loss was determined in Professor Burstall's experiments.

'Professor Burstall commented on his (Professor Burstall's) results as follows: "This result does not accord with the usual belief that economy increases with compression, when a suitable mixture is used. The cooling action of the walls, however, affects the result materially. Consider the contents of the cylinder at the end of compression. The gas is confined in a space 16 ins. in diameter, at the highest compressions about $3\frac{1}{4}$ ins. long, and at the lowest compressions about $6\frac{1}{2}$ ins. long, the gas being entirely surrounded by water-cooled surfaces. This being the case, the leakage of heat during compression will be greater proportionately at the high than at the low compressions, because the higher compression is accompanied by a higher density and by a temperature difference between the walls and the charge, and this more than compensates for the reduction of the area of surface exposed to the gases."'

Here Professor Burstall appeared to attribute loss in economy at high compressions to loss of heat during compression. In this the writer differed entirely from him. Loss of heat during compression changed efficiency very slightly indeed. The loss of efficiency caused by using isothermal compression instead of adiabatic was too small to be noticed in such experiments as those of Professor Burstall. Possibly Professor Burstall might have meant this paragraph to apply to the cooling action of the walls, as affecting the flame enclosed within them after explosion. If so, then his experiments appeared to show that the fall in efficiency which he considered his experiments to have proved was not due to loss of heat. As the writer had pointed out, the mean loss to cooling water shown by experiment C was 20·6 per cent. of the whole heat, and experiment D 19·4 per cent. Now the mean loss in experiment B was only 17·2 per cent., and in A 22·2 per cent. The great fall in the dotted line 2 through Burstall's experiments occurred between C and B, and yet in C the heat loss to walls was 20·6 per cent., and in B only 17·2. The fall of efficiency could not be due to increased heat loss. It was true that in the A experiments the heat loss had increased slightly to 22·2 and 20·6, but this could only account for a very small relative fall in efficiency. The heat loss numbers did not support Professor Burstall's contention.

In his view, Professor Burstall's experiments, when properly read, supplied a most gratifying corroboration of the accuracy of the results

deduced by the Committee of the Institution of Civil Engineers, as his experiments closely followed the air standard curve marked 3. They also very closely followed the curve deduced by the writer from the actual properties of the working fluid, marked 5. It should be noted that the curves marked 4 and 5 were calculated, including values of apparent increasing specific heat, as determined by him in 1906. He trusted it would be fully understood that although he differed from Professor Burstall in some of his conclusions, he considered his work to corroborate in the most valuable way the correctness of the deductions generally accepted by practical gas-engine constructors. His numbers would prove very useful to future investigators.

In the interesting discussion which followed the reading of Professor Burstall's report at the Institution of Mechanical Engineers, Captain Sankey and Professor Hopkinson pointed out that efficiency must increase slowly with the higher compressions, and Captain Sankey stated:

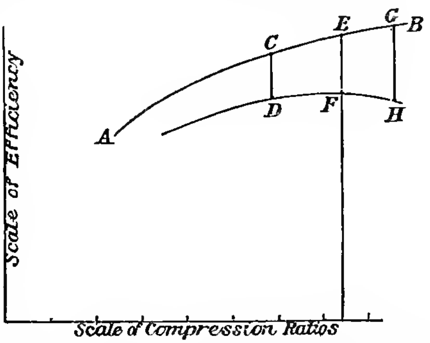

FIG. 119.—Curves showing increase of thermal efficiency and increase of heat loss with increasing compression. (*Sankey*)

'It appeared to him that the report boiled down to two important facts: one, the reduction of thermal efficiency after the certain ratio of compression had been reached, while the other was the fact that with moderate mean pressures the best results were obtained. Possibly the first point might be explained in a simple manner. Assuming variable specific heat, the thermal efficiency of an ideal gas engine would be represented by the curve A, B, fig. 119, which rose quickly at first, but after a time rose more slowly. With moderate compressions and moderate temperatures in the cylinder the losses would not be very great, and might be represented by the progressive distances C D, E F, and G H. Joining the points D, F, H, there would then be obtained the thermal efficiency of the actual engine rising quickly, and then, after a certain point, falling down, the point of maximum thermal efficiency being at F, which, for the C trials of this particular engine, was at about 7·2. The foregoing might perhaps explain the results that Professor Burstall had obtained experimentally. At the Derby trials of suction plants, made about eighteen months ago, there were a great number of small engines which had very varying compressions, and although in that case the efficiency of the producer itself came into

consideration, it was quite noticeable that after a certain compression the final result was not so good. One engine had a very high compression indeed, and the result of that engine—otherwise an excellent engine—was not so good as that of another engine with a lower compression.'

This is quite true; the loss of efficiency might be explained exactly as Captain Sankey puts it, and no doubt a point would arise when compression was raised to such intensity that the efficiency curve would behave in the manner Captain Sankey suggests. But the internal proof appears to be against the assumption that such changes really occurred in the Burstall experiments.

Professor Hopkinson shares the view held by the author, as will be seen from his remarks:

'Professor Bertram Hopkinson said that Captain Sankey had drawn attention to two points which he considered of special importance, and the speaker would like to deal somewhat further with both of these points. The first related to the conclusion that at a certain point the increase of compression in the engine under discussion did not lead to any increased efficiency. One of the best established conclusions in gas-engine theory and practice was that in an engine working on the gas-engine cycle, in which the mixture was compressed, and, after firing, was expanded to the extent to which it had previously been compressed, the efficiency went up with the compression; moreover, that it went up more or less in proportion to the efficiency of an ideal air engine working to the same compression. Professor Burstall's experiments at the lower compressions added further proof—if any were needed—of that belief. These experiments showed that from as low as 80 lbs. up to a compression of about 180 lbs. the efficiency increased. Professor Burstall had, however, inferred from experiments at higher compressions that the efficiency did not increase any further, and that therefore the ratio fell—a result that did not agree with the usual belief, that economy increased with compression. The evidence put forward did not appear to the speaker to establish Professor Burstall's conclusions. On comparing the C trials with the A trials, it would be seen that the C trials corresponded to a compression ratio of 7·22. Worked out according to the usual formula, $1 - \left(\frac{1}{r}\right)^{0·4}$, the air-cycle efficiency for the compression ratio of 7·22 would be 0·543. In the case of the A trials, with a compression ratio of 8·07, the corresponding value of the efficiency would be 0·566. On the basis, therefore, of the theory that the gas-engine efficiency went up in proportion to the air-cyle efficiency, no great increase in efficiency as between the A and C trials would be expected. The difference was only about 4 per cent. on the amount of work done for a given charge of gas. The

speaker felt doubtful whether by the methods which Professor Burstall employed it would be possible to detect any such increase of efficiency as that. He did not know exactly the degree of accuracy which Professor Burstall believed himself to have obtained. He would hardly suppose that the author would swear to his indicator diagrams being accurate within 5 per cent., and if that were so, the results obtained—0·415 for the C trials and 0·394 for the A trials—might be interpreted as showing that in this engine the efficiency was substantially the same for the C and the A trials. It had been stated in the paper that other evidence would be forthcoming, but from the evidence as it stood it was not possible to infer that there was really any departure from the law—it might be so described—that efficiency went up with compression. Certainly the evidence was not conclusive.

'The other point that Professor Burstall had dealt with, and which had been brought out very clearly by the results, was that maximum efficiency corresponded with comparatively weak mixtures, and therefore with low mean pressure. That, speaking generally, was well known; anyone working with a gas engine knew that the best efficiencies were not obtained with the heaviest charges. The speaker had himself made experiments upon the point, the results of which he had given in his paper read before the Institution last October. In that paper it was shown that with the particular engine dealt with the efficiency with a charge of $\frac{1}{10}$ of a cubic foot of coal gas per explosion was about 37 per cent., and that with a charge of 0·13 cubic foot per explosion the efficiency fell to 33 per cent.'

On the question, too, of the effect of loss of heat during compression, Professor Hopkinson's view is the same as that of the author:

'Professor Burstall had, said Professor Hopkinson, made a remark which might possibly have been a mistake. Speaking of the effect of higher compression in possibly reducing efficiency, the author had said: "This being the case, the leakage of heat during compression will be greater proportionately at the high than at the low compressions, because the higher compression was accompanied by a higher density and by a higher temperature difference between the walls and the charge, and this more than compensates for a reduction of the area of surface exposed to the gas." That rather indicated that, in Professor Burstall's view, the lower efficiency which he thought was obtained by increasing the compression was due in some way to the greater loss of heat during the compression of the charge—that was, before explosion. The speaker did not consider there was anything in that; the effect of loss of heat during the compression of the charge was really very small. Anyone could convince himself of that by working out the efficiency of an ideal air-cycle, in which, instead of compressing the gases adiabatically, they were compressed isothermally, and then,

with the same rise of temperature, expanded adiabatically, the whole of the heat developed by the compression of the gas being lost. It was surprising what a small difference that change in the cycle made in the efficiency. The reason was that, though heat was lost during compression, less work was done in compression, equalising the loss. Possibly what Professor Burstall really meant was that the leakage of heat during expansion was greater proportionally at the higher than at the lower compressions, which was no doubt true for very high compressions. The effect of these heat losses, occurring in an engine of the size of that under consideration, upon thermal efficiency was, however, surprisingly small. If the heat loss were totally suppressed, it would make very little difference. Mr. Dugald Clerk had clearly shown this some time ago before the Institution of Civil Engineers. He did not think that Professor Burstall's figures quite sustained the view that the reduced heat loss was the cause of the supposed higher efficiency with the lower compression. In the B trials the average heat loss was 17 per cent., and the thermal efficiency was 0·398, whereas in the C trials the thermal efficiency was 0·415, and in spite of their higher thermal efficiency there was also a bigger heat loss on the average of the four trials.'

Professor Burstall's experiments on heat absorbed in the various water jackets are most interesting, and the estimation of the heat carried away in the separate water systems of breech, barrel, piston, and valves will enable us to obtain values of a most useful kind.

Taking Burstall's broad division of the total heat going to the jackets as given in the paper :

Breech	1·0
Barrel	1·5
Valves	0·75
Piston	0·5
	3·75

we get the following percentage division of the heat carried away in the jackets :

	Per cent.
Breech plus valves	46·5
Barrel	40·0
Piston	13·5
	100·0

The piston area is practically equal to the area presented by the breech and valve surface (see fig. 115), and if the valves never opened there is no reason why the breech and valve surface should absorb more heat than the piston surface. But, as the exhaust valve does open, the highly heated exhaust gases rush past the valve and impinge upon the curved discharge passage leading to the exhaust pipe; a large

additional quantity of heat thus flows into the exhaust valve and into the water jacket of the exhaust discharge passage, which forms part of the breech jacket, so that the heat flow into that also increases. The exhaust gases, also discharged at atmospheric pressure on the return movement of the piston, pass round the exhaust valve and impinge on the jacket. If, however, we deduct the piston heat, 13·5 per cent., from the breech and valve heat, 46·5 per cent., that is, $46{\cdot}5 - 13{\cdot}5 = 33$ per cent., we obtain the proportion of jacket heat which has come from this impingement of the exhaust air valve and passage as it leaves the engine.

Here we have a means of correcting or corroborating the conclusions arrived at earlier in this discussion. The J trials of Burstall have nearly the same compression ratio as the Institution of Civil Engineers' Committee trials with the 'X' engine: Burstall's J $\frac{1}{r} = \frac{1}{5{\cdot}45}$; 'X' engine $\frac{1}{r} = \frac{1}{5{\cdot}35}$. Burstall's indicated thermal efficiency is 34·5 per cent., 'X' engine 34·7 per cent.; the efficiencies are practically equal; the mean heat flow to the jackets, Burstall, 28·6 per cent., is the proportion of the total heat given to the engine, and of this one third, 33 per cent., passes into the exhaust valve and passage. That is, $\frac{28{\cdot}6 \text{ per cent.}}{3} = 9{\cdot}5$ per cent. of the total heat appears as expansion loss, when it is really incurred after the expansion is complete.

In the 'X' engine Committee trial, the heat flow into the jacket is 25·4 per cent. and $\frac{25{\cdot}4}{3} = 8{\cdot}5$ per cent. That is, if the 'X' engine valve and passage absorbed the same proportion as Burstall's J experiment, the true heat flow to the cylinder walls during the explosion and expansion stroke should be

$$25{\cdot}4 - 8{\cdot}5 = 16{\cdot}9.$$

This approximates closely to the actual heat flow given by Clerk's new diagram trial of the 'X' engine, and corroborates the correctness of the balance-sheet on pp. 270 and 273.

It is unfortunate that Professor Burstall's experiments were not complete enough to supply data for a heat balance-sheet. Professor Burstall, in the author's view, relied too much upon the accuracy of the indicator, and attached far too small importance to brake tests and other means of correcting or corroborating the conclusions arrived at by the indicator. It was intended by the Gas Engine Research Committee that all these controlling means should be used, but unfortunately the difficulties of working the engine at very high compressions, together with the varying quality of the Mond gas at the light load on the large 500 HP producer, introduced troubles of a

practical nature which fully occupied Professor Burstall's attention. Those difficulties detracted from the value of Burstall's investigation from the scientific point of view, but properly used Burstall's experiments afford valuable corroboration of the work of other investigators, as they are the only tests conducted through such a long range of compression values with producer gas.

The Standards Committee of the Institution of Civil Engineers, Professor Hopkinson's investigations, and the present writer's investigations have been made with full appreciation of indicator difficulties, and, accordingly, every care has been expended upon the corroborative experiments of brake, air measurement, exhaust calorimeter, and heat flow by measurement in jackets and by the author's new diagram method.

Professor Burstall, however, gives some numbers which enable the mechanical efficiency of the engine to be valued.

He determined the frictional and fluid losses of the engine to be 22 HP.

The mean values of the IHP for the A, C, D, F, and J trials are respectively 84·7, 95·1, 101, 104, 95·4, and, assuming 22 HP to be required to drive the engine in each case, we have the following values of mechanical efficiency, indicated thermal efficiency, and brake thermal efficiency.

	Trials				
	A	C	D	F	J
	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.
Mechanical efficiency	74	77	78	79	77
Indicated thermal efficiency	39·4	41·5	40·4	35·9	34·5
Brake thermal efficiency	29·2	32	31·6	30·8	26·6

The mechanical efficiency of this engine thus apparently varies between 74 per cent. to 79 per cent., and the brake thermal efficiency does not exceed the values obtained for smaller engines at similar compression ratios by the Institution of Civil Engineers' Committee and Professor Hopkinson.

It might be thought that the mechanical efficiency of the modified Premier engine was lower than that of the normal pressure engine, but an examination of Humphrey's test of a 500 HP Premier engine in 1900 shows the mechanical efficiency to be 81·2 per cent., so that the above numbers do not appear to be far out.

A test made by Professor Burstall on November 25, 1904, with a Crossley engine of 14 ins. diameter cylinder and 21 ins. stroke, using injected water to keep down pre-ignition, gave the following results:

	Per cent.
Mechanical efficiency	82·2
Indicated thermal efficiency	37·43
Brake thermal efficiency	30·8

It will be observed that the mechanical efficiency is low and the indicated thermal efficiency high, while the brake thermal efficiency is practically the same as that of the National Company's 'X' engine, 30·8, against 29·9 per cent., notwithstanding the fact that the compression ratio of the Crossley engine is $\frac{1}{8{\cdot}7}$, while that of the National is only $\frac{1}{5{\cdot}34}$. In this case little gain has been obtained by the higher compression.

It is improbable, however, that Burstall's values for mechanical and indicated thermal efficiency are correct, as is shown by the following test of another Crossley engine of $11\frac{1}{2}$ ins. diameter and 21 ins. stroke :

	Per cent.
Mechanical efficiency	86·6
Indicated thermal efficiency	35·3
Brake thermal efficiency	30·6

Here we have a lower indicated efficiency, the same brake efficiency, and a higher mechanical efficiency.

Thermal and Mechanical Efficiency of the Diesel Engine

Very high indicated thermal efficiencies are claimed for the Diesel engine, but in the author's view the term 'IHP' has not been properly defined with regard to this engine. Indicated power on an engine of any cycle of operations is the total positive work performed by the working fluid upon the piston minus the work performed by the piston on the working fluid—that is, the negative work. The negative work may be defined as work done by the piston on the working fluid necessary for the thermodynamics of the cycle; it is work which is done by the piston to be returned again on a following operation. This negative work must be carefully distinguished from the fluid resistance incidental to charging the cylinders; such resistances are not reversible—that is, they form part of the mechanical loss of the engine; they are not recoverable, but they may be reduced indefinitely by increasing the dimensions of valves and passages. It would be improper, for example, to give as the indicated power of an Otto cycle engine the area of the total diagram on the power stroke without deducting the negative work incurred on the immediately preceding compressing stroke; yet something of this nature is often done in dealing with Diesel engine diagrams. The negative work of the pumps

compressing the high-pressure air to force in the oil in a state of spray is sometimes treated as a frictional loss, whereas it is really negative work, which in the main disappears in the pump to reappear in the power cylinder.

For this reason Diesel engine tests often show very low mechanical efficiencies. Take, for example, the following tests of a 500 HP Diesel engine made by Mr. Longridge :

TESTS OF A 500 B H P DIESEL OIL ENGINE, BY MICHAEL LONGRIDGE, M.INST.C.E.

Three cylinders : diameter, 22·05 *ins.* ; *stroke,* 29·52 *ins. February* 1905

Load	Full	Half	Light
Duration of test . . . . . minutes	115·25	129	58
Revolutions per minute	152·8	150·3	150·2
Jacket water per minute	169·85	157·85	140
Rise of temperature of jacket water . .	81°·1 F.	58°·3 F.	35°·8 F.
Temperature of exhaust gases . . .	806° F.	496° F.	275° F.
Analysis of exhaust gases . . . CO_2	6·8	3·2	
N	51·3	21·5	
Air	41·9	75·3	
Oil used per hour . . . . pounds	207·2	102·8	45·76
Blast pressure . . . . atmospheres	66·3	50·9	35
Maximum pressure by indicator diagrams, pounds per square inch	525	505	500
Average mean effective pressure, pounds per square inch	96·7	56·33	25·4
Indicated horse-power	634·8	353·6	163·3
Oil per indicated horse-power per hour, lbs.	0·3264	0·2828	0·280
Brake horse-power	562·5	245	54·6
Horse-power absorbed in friction .	132·3	118·6	108·7
$\dfrac{\text{Brake horse-power}}{\text{Indicated horse-power}}$	0·805	0·675	0·334
Oil per brake horse-power hour . pounds	0·4123	0·4196	0·838
Indicated horse-power in compressor cylinders	40	28·8	18·2
Estimated brake horse-power of engine if pumps driven by engine	458·7	213·8	32·4
Estimated mechanical efficiency if pump had been driven by engine	0·723	0·588	0·198
Oil per brake horse-power hour if pump driven by engine	0·451	0·481	1·415

Here the indicated HP at full load is given as 634·8 and the mechanical efficiency 80·5 per cent., so that the brake HP must be 634·8 × 0·805 = 510 ; but it appears that the air-pressure and oil-pressure pumps have been driven independently of the engine, so that the estimated brake HP of the engine with pumps driven by the engine is 458·7 and the mechanical efficiency on this basis is only 72·3 per cent., and the oil consumed per BHP 0·451 lbs.

Taking the oil as having a lower calorific value of 18,000 B.Th.U. per lb., this gives a brake thermal efficiency of 31·7 per cent. Now on

this basis the indicated thermal efficiency would be $\frac{31 \cdot 7}{0 \cdot 723} = 43 \cdot 8$ per cent.—that is, an indicated thermal efficiency of 43·8 per cent.

The power to drive the pumps is $510 - 458 \cdot 7 = 51 \cdot 3$ horse, but of this only 40 is indicated in the compressor cylinders, so that the legitimate indicated HP should be, say, $635 - 40 = 595$, and the mechanical efficiency $\frac{459}{595} = 0 \cdot 77$. That is, the real mechanical efficiency of the engine is 77 per cent., and on this basis the indicated thermal efficiency is $\frac{31 \cdot 7}{77} = 41$ per cent.

For this 500 HP engine at full load on this test, it may be taken that the following values more truly represent the results in the sense defined :

Indicated power	595 horse.
Brake power	459 ,,
Mechanical efficiency	77 per cent.
Indicated thermal efficiency	41 ,,
Brake thermal efficiency	31·7 ,,

The compression ratio is not given in the particulars, so that the efficiency ratio relative to the air standard cannot be given.

The following particulars have been calculated by the author from the test of a Diesel oil engine by Mr. Ade Clarke : Compression ratio $\frac{1}{r} = \frac{1}{14 \cdot 29}$, for which the air standard value is 0·655.

The experimental absolute efficiency, calculated on the indicated horse-power, in one test was 39·9 per cent., so that the efficiency ratio was

$$\frac{0 \cdot 399}{0 \cdot 655} = 0 \cdot 61 \text{ nearly.}$$

The experimental absolute efficiency, calculated on the brake horse-power was 31·2, so that the efficiency ratio was 0·476. The following are the values :

	Per cent.
Mechanical efficiency	77·6
Indicated thermal efficiency	39·9
Brake thermal efficiency	31·2

The results of the two tests are very similar ; broadly, an indicated thermal efficiency of 40 per cent. may be expected from these engines, but the mechanical efficiency is so low, about 77 per cent., that not more than 31 per cent. can be expected for the brake thermal efficiency.

Thermal Efficiency of the Two-cycle Engines

Otto's method is probably the readiest and easiest solution of the problem of attaining in a practicable manner the advantages of com-

pression ; in some points, however, the advantages are accompanied with compensating disadvantages.

Only one impulse for every two revolutions is obtained ; the engine is therefore stronger and heavier than need be if impulse at every revolution were possible. It is also more irregular in its action than more frequent impulses would give.

The Clerk engine was invented by the author with the view of obtaining impulse at every revolution, while getting at the same time the economy due to compression.

At first blush it seems a very simple matter to make a compression gas engine to give an impulse for every revolution ; this was the author's opinion when he commenced work for the first time upon gas engines using compression in October 1876. Since then he has had occasion to modify the opinion : the difficulties are very great ; any engineer who doubts this will speedily be convinced upon making the attempt.

It was not till the end of 1880 that the author succeeded in producing the Clerk engine ; before that time he had several experimental engines under trial, one of which was exhibited at the Royal Agricultural Society's show at Kilburn in July 1879. This engine was identical with the Lenoir in idea, but with separate compression and a novel system of ignition.

The Clerk engine was the first to succeed in introducing compression of this type, combined with ignition at every revolution ; many attempts had previously been made by other inventors, including Mr. Otto and the Messrs. Crossley, but all had failed in producing a marketable engine.

In the Clerk engine the whole cycle is completed in one revolution and an impulse given to the crank on every forward stroke of the piston, when working at full power.

The engine contains two cylinders, one for producing power, the other for taking in the combustible charge and transferring it to the power cylinder. At the end of the motor cylinder is left a compression space of a conical shape, and communicating with the charging or displacing cylinder by a large automatic lift-valve opening into the space ; at the other end of the cylinder are placed V-shaped ports opening to the atmosphere by the exhaust pipe ; the motor piston, when near its outer limit, overruns these ports and allows the cylinder to discharge. The pistons are connected in the usual manner by connecting rods, the motor to the main crank of the engine, the displacer to a crank pin in one of the arms of the flywheel ; the displacer crank is in advance of the motor crank, in the direction of motion of the engine, by a right angle. The displacer piston on its forward movement takes in its charge of gas and air, and has returned a fraction of its stroke when the

motor piston uncovers the exhaust ports. While crossing the centre, opening and shutting these ports the displacer piston has moved in almost to the end of its cylinder, discharging its contents into the space and forcing out at the exhaust ports the products of the previous ignition. The proportions of the two cylinders are so arranged that the exhaust is as completely as possible expelled, and replaced by a cool explosive mixture, which thoroughly mixes with any exhaust remaining, cooling it also to a considerable extent. Care must be taken in the arrangement of the parts that an excessive volume is not sent from the displacer, otherwise it may reach the exhaust ports and gas discharge unburned.

The return stroke of the motor piston now compresses the mixed gases, and, when at the extreme end, the igniting valve fires the mixture, the piston moves forward under the pressure thereby produced, till the opening of the exhaust ports causes discharge and replacement as before. In this way an impulse is given at every revolution, and the motive power applied to greater advantage. The motor cylinder is surrounded by water for cooling, but this is unnecessary with the displacer, as it uses only cool gases. The pressures used are high, so that both motor piston and its connections are made very strong; the pressure on the displacer piston is very little, so the connections are light. It is not a compressing pump, and is not intended to compress before introduction into the motor, but merely to exercise force enough to pass the gases through the lift valve into the motor cylinder, and thus displace the burned gases, discharging them into the exhaust pipe. The pressure to be overcome is only that due to resistance in the exhaust passages and pipes and the lift valve.

In the Clerk cycle engines the considerations which govern power and economy very closely resemble those of the Otto cycle, but there are several points which require to be carefully considered—points, indeed, of considerable difficulty—with regard to which the Otto cycle is a far easier cycle than the Clerk cycle. In the Clerk cycle, the charging has to be accomplished in the motor cylinder, while the crank is passing through an angle of about 80°. Sometimes a little larger angle is allowed, but, roughly, 80° of the crank angle is the limit for charging of the motor cylinder. Because of this, much larger inlet valves and very much larger discharge areas are required in the Clerk than in the Otto cycle. In the Otto cycle the charging stroke occupies not only the whole of one stroke, which amounts to 180° of the crank movement, but in addition a further 40°, which permits the inlet valve to be held open considerably over the out centre, and also to be opened a little before the centre on the in-stroke. In consequence, the Otto type of engines allows three times the time interval to charge the

cylinder, at a given rate of revolution, that it is possible to allow in the Clerk cycle. For a given valve area, the velocity of charge entrance in the Otto cycle is about a third of that in the Clerk cycle. This means that the Clerk cycle engines are more difficult to charge, and require greater power expenditure to charge the cylinder than the Otto cycle. That is the great weakness of the Clerk cycle as constructed at present.

Then there is this further point. In the Otto engine, when the piston moves out, taking in its charge, there is no question of any possible discharge at the exhaust ports, because the piston is sucking in the charge by a partial slight deficit of atmospheric pressure, and there is no exhaust port open through which fresh charge may be lost. In the Clerk cycle, the proportions of displacer and motor cylinder have to be very accurately ascertained, otherwise part of the charge entering the cylinder is apt to pass right down through the centre of the exhaust gases which are being displaced, and pass out of the exhaust port. That was overcome in the early Clerk engines by the peculiar conical shape of cylinder end, which has been since consistently adhered to in all the engines operating on the Clerk cycle. That difficulty, however, is met in two ways. One way is to put in a smaller charge than in the Otto cycle, but that has the disadvantage of leaving too much exhaust gas and also giving a smaller power of engine. Consequently, every designer of the Clerk cycle engine attempts to get in the full charge. The best method is to send into the cylinder, first a good heavy charge of air to displace the exhaust products, and then to follow it with a somewhat strong charge of gas and air. That is what was done by the author in 1881, and that is what is being done to-day in all the large gas engines. That, however, is a somewhat difficult thing to do. The consequence is that if the cylinder be charged as fully as it would be in an Otto cycle, a slight proportion of gas is lost at the exhaust ports, and although in a small engine with a comparatively light load the economy very closely approaches the best Otto economy, yet the maximum efficiencies that are possible with the Otto cycle have not been obtained with any two cycle engines. Because of this, the Clerk cycle has been reserved for somewhat special uses. It has come into extensive use of recent years in connection with large blast-furnace gas engines.

The modern representative of the Clerk cycle engine is known as the 'Koerting' engine, a German production, designed and built by the well-known engineers Messrs. Koerting, of Hanover; but, so far as its essentials are concerned, it was invented in England twenty-eight years ago. The charge is admitted at the inlet valves placed at the conical ends of the cylinder, and instead of having one pump for gas and air mixed, two pumps are provided, one of which is for gas and the

other for air. This engine, instead of being single-acting as the Clerk engine was, is double-acting, so that, so far as the main crank is concerned, there are two impulses every revolution just as there are in a steam engine.

In this Koerting engine the piston is in the middle position and does not overrun the exhaust ports till the end of its stroke, when it allows the pressure in the cylinder to fall to that of atmosphere.

Then, immediately the pressure has fallen to atmosphere (this requiring about 20° movement of the crank), the charge inlet valve is opened and air is first pumped in from the air-pump. After the air has been flowing in for some little time to throw the exhaust gases forward, the gas is pumped in from the same valve, so that no mixture is made until the gas and air mixture enters the cylinder. That is a very important point. In a small engine the gas and air may be mixed in the pump, but in a large engine the charge of gas and air must be mixed just as it enters the cylinder. In a small engine an occasional back ignition into the displacer is no great matter; in a large engine it will be a serious matter; so that in the large engine the gas and air are kept separate until they flow into the motor cylinder. When the charge has entered, air first and then gas and air, and has displaced the exhaust products, the piston is closing the ports which shut off the exhaust after a movement of about 40° of the crank; the compression then proceeds, and ignition and expansion take place, so that an impulse is obtained at every forward stroke behind the piston, and at every backward stroke in front of the piston.

The Oechelhauser engine resembles the Clerk and Koerting engines in discharging its exhaust through ports in the cylinder which are overrun by the piston; it has two opposing pistons, driven from a crank in the front part of the engine; there are three crank pins, one the main crank pin driving the front piston; then outside of that there are two other crank pins, disposed exactly opposite—that is, 180° from the main crank. These two opposite pins drive the second piston by separate side rods and cross-head, and the two pistons therefore move out at equal velocity. This arrangement produces a very nicely balanced engine, all the strain of the explosion being removed from the frame and taken by the cranks only. The two pistons overrun the ports, the one at one end and the other at the other. In the front end, the piston overruns the exhaust ports, and the pressure falls to atmosphere exactly as it does in the Clerk and the Koerting engines. A little later the back piston overruns the air ports. The air is accumulated in a chamber by means of a pumping arrangement, and this air first passes into the cylinder from ports placed all round it, driving out the exhaust gases. In this way the cylinder is cleared of the hot incandescent matter before the gas enters. Then as the piston moves

further back the next ports are uncovered, and the gas enters, coming from a separate reservoir. The mixture of gas and air flows right up the cylinder, and discharges the exhaust gases and the air that has been put in first; then compression takes place, and then ignition, expansion, and so on, just as in the other cycles. This engine has one great advantage over other forms of engine working on the Clerk cycle, viz. that there are no valves, so far as the combustion and explosion are concerned, except the pistons themselves. The two pistons form the whole of the valve arrangements, and no pressure of explosion can come on any other valve. There are no lift valves exposed to the consequences of the explosion, so that a very large valve area may be allowed. The engine, however, has disadvantages. It is a more expensive engine to build, and a very considerable weight of metal is required for a given power, but it is a very smooth-running engine.

The thermodynamic considerations relating to the two-cycle engines do not differ in any way from those proper to the four-cycle engine, although the practical difficulties of obtaining high indicated thermal and brake efficiencies are greater, for the reasons which have been stated.

Like the four-cycle, however, the two-cycle engine has greatly advanced in economy since 1884, as will be seen from the following table:

INDICATED AND BRAKE THERMAL EFFICIENCY OF TWO-CYCLE ENGINES FROM 1884 TO 1908

Mechanical efficiency	Name of experimenter	Year	Dimensions of motor cylinders	Indicated thermal efficiency	Brake thermal efficiency	Type of engine
Per cent.				Per cent.	Per cent.	
84	Garrett	1884	9″ diam. × 20″ stroke	16·4	14	Clerk-Sterne
—	Stockport Co.	1884	—	—	11·2	Andrews & Co.
83	Clerk	1887	9″ diam. × 15″ stroke	20·2	16·9	Clerk-Tangye
—	Atkinson	1885	7½″ diam.	—	15	Atkinson
75	Meyer	1903	26⅝″ × (2″ × 37½″)	38	29	Oechelhauser
75	Mather & Platt	1907	—	30·6	23	Koerting

From these values it will be seen that the two-cycle engine has, like the four-cycle, greatly improved in both indicated and brake thermal efficiency, rising from 16·4 per cent. in 1884 to 38 per cent. in 1903 for the former, and from 14 per cent. to 29 per cent. for the latter.

It is to be remembered, however, that the numbers in this table cannot be accepted as having the degree of accuracy attained in the four-cycle tests. The thermal efficiencies for the earlier engines have been calculated on the assumption that the calorific value of Glasgow and Birmingham gas is correctly given by earlier analysis of the coal gas. The calorific value of the gas was not determined at the time.

Determinations of calorific value were made, however, by Meyer and Mather & Platt.

It will be noted that the mechanical efficiency of the Koerting and Oechelhauser engines is low, only 75 per cent., and in this respect the early Clerk engine was undoubtedly more successful with 84 per cent.

Discussion of Differences between Actual and Ideal Engines of the Third Type

In the present and the preceding chapters V., VI., VII., and VIII. ample experimental data have been given to enable us to discuss more fully the several ways in which the actual engine departs from the ideal as mentioned in Chapter IV. The eight points there raised shall now be dealt with in order.

According to—

(1) The working fluid loses heat to the walls enclosing it after its temperature has been raised to the highest point; . . .

The foregoing experiments have shown that in a National gas engine, with a cylinder of 14 ins. diameter and 22 ins. stroke, running under ordinary conditions of full load, this loss amounts to only 16 per cent. of the heat given to the engine, and in a Crossley engine of $11\frac{1}{2}$ ins. cylinder by 21 ins. stroke, under similar conditions, the loss is only 21 per cent.

The experiments upon explosions in closed vessels make it very probable that, given similar vessels of differing dimensions—cubes, spheres, or cylinders—heat loss is inversely proportional to the side of the cube, the diameter of the sphere, or the cylinder.

Assume, therefore, similar conditions between these two engines as to density, mean temperature, and engine revolutions; if the heat loss in the 14-in. cylinder be 16 per cent., in the $11\frac{1}{2}$-in. cylinder it should be $\frac{14 \times 16}{11 \cdot 5}$ = say, 19·5 per cent. The density in the Crossley engine is greater, and the surface exposed also greater proportionally, so that the approximation is sufficiently good.

It must not be supposed that the heat loss increases in this ratio without regard to other circumstances, otherwise small petrol engines would lose the greater part of their heat; it depends also on revolutions per minute, and, other matters being equal, it is proportional to the times of exposure.

The division of the loss during the stroke has also been shown by the author's experiments for the same National engine to be 50 per cent. on the first $\frac{3}{10}$ stroke and 50 per cent. in the last $\frac{7}{10}$ stroke. Heat loss can be rendered practically negligible by increasing dimensions.

The increased mechanical work to be obtained, however, by

suppressing entirely a 20 per cent. loss is but small when the effect of heat flow throughout the stroke is considered.

The first cause of loss, so far as thermal efficiencies are concerned, possesses but little practical importance in engines of moderate cylinder dimensions.

(2) The working fluid often gains heat when entering the cylinder at a time when it should remain at the lowest temperature ; . . .

The experiments prove that in the four-cycle engine this cause has little effect on indicated thermal efficiency. The usual suction temperature in a well-designed engine, running under nearly full load, does not exceed 100° C., and this initial temperature is not harmful.

The power obtained per cylinder is reduced by heating of the entering charge, because the total charge weight is reduced. Power obtainable is proportional to charge weight, so that, for an atmospheric temperature of 20° C., greater power could be obtained than for 100° C. The power is proportional to absolute temperature, so

$$\text{Power possible at } 20^\circ \text{ C.} = \frac{\text{power at } 100^\circ \times 373}{293}.$$

From the point of view of power, the experiments prove that the colder the charge before compression the greater the power obtained.

(3) The supply of heat is never added instantaneously, as required in some types.

The experiments show that explosion in closed vessels, whether at one or several atmospheres, at temperatures of about 20° C. or so, takes place slowly for such dilutions as are most useful in gas engines.

In the author's experiments, for example, with Oldham gas, 1 gas 9 air, fired at atmospheric pressure, the time of explosion was 0·08 second ; in Bairstow and Alexander's experiments, with a rich mixture of 1 gas 7 air at initial pressure 35 lbs. absolute, the time of explosion was 0·105 second. These times are too long for effective use in an engine running even at the moderate speed of 160 revolutions per minute.

But even weak mixtures ignited under compression in an engine cylinder have times of explosion shorter than this. The diagrams, fig. 105, from the 'X' engine show times of explosion of about $\frac{1}{48}$ second, while Hopkinson's diagrams, fig. 108, show even shorter periods.

The heat of compression and the movement of the charge in the cylinder give sufficiently rapid rates of ignition with mixtures as weak as 1 gas to 11 air.

It must not therefore be supposed that the time of explosion of a cool mixture in a closed vessel is a guide as to the time in the actual engine cylinder. The diagrams referred to show that up to such temperatures as 1500° C. to 2000° the rate of heat addition may be made as close as is desirable to the ideal of instantaneous addition.

(4) The working fluid is not the pure dry air assumed in Chapter III.; it is a mixture of nitrogen, carbonic acid, steam, and oxygen, with a different specific heat from air, which specific heat increases considerably with temperature.

The experiments prove that the specific heat of the working fluid of the gas engine increases with increase of temperature; but the increase of specific heat does not account for all the phenomena of the working stroke. Combustion is not complete at maximum temperature, but extends some way on in the expansion stroke, and in some cases even into the exhaust stroke.

The separation of the different actions is by no means complete, and even in determinations of specific heat, such as those of Holborn and Austen, and Holborn and Henning, where Regnault's calorimetric method is adopted, it is possible, according to Callendar, that systematic errors may exist of as much as 10 per cent. on the total energy from the chosen zero-point.

The explosion experiments, however, of Mallard and Le Chatelier, Langen, Hopkinson, and the author make it possible to determine *apparent* specific heat values—that is, values which include other things than specific heat change. Tables of such specific heats for the working fluid of the 'X' engine are given at p. 235.

Hopkinson has prepared a curve of total energy, which is given at fig. 113, from the results of all the observers just mentioned. By means of his own experimental values the author has come to the conclusion that the ideal efficiency of the actual working fluid of the 'X' engine is, broadly, 20 per cent. lower than it would have been had air at constant specific heat been used in the supposed perfect engine. The air standard efficiency for that engine would have been 49 per cent., and the actual fluid standard is really 39·5 per cent. The engine realises 88 per cent. of its possible efficiency with its actual working fluid, instead of only 71 per cent., as would be the case with air. Hopkinson's results corroborate those of the author. He finds that the 40 HP Crossley engine with which he experimented has an air standard efficiency of 52·2 per cent., and for a weak mixture the actual fluid standard is 42·4 per cent., and that the Crossley engine realises 87 per cent. of its possible efficiency. For a rich mixture Hopkinson finds that the actual fluid standard falls to 39·4 per cent., and the engine realises 83 per cent. of its possible efficiency.

Many investigators are now at work in Britain and on the Continent on the more exact determination of the physical and chemical constants of the working fluid, and the report of a British Association Committee on Gaseous Explosions will be found in the Appendix, which deals fully with the subject. The thermodynamics of the working fluid, however, are now understood as they never were before.

(5) Combustion is not complete when the maximum temperature of the working fluid is attained, and in some cases it is not complete when the exhaust valve opens.

The method of apparent specific heat summarised under (4) includes the effect of such imperfect combustion as usually occurs in a gas engine having good mixture and excess of air, but it does not include cases where mixture is bad and sometimes inflammable material is present in excess. This occurs very often in petrol engines, and the Automobile Club experiments, those of Hopkinson and the author, prove that sometimes as much as 30 per cent. of the heat possible from the petrol may be discharged from the engine in the form of carbonic oxide, hydrogen, and hydrocarbon. As much as 10 per cent. of carbonic oxide has been found in the exhaust gases of a petrol engine. The oil engine generally discharges more of its fuel without burning it than does the gas engine. The difficulty arises from the nature of the operation of carburetting and the absence of true proportionality between air and petrol under the varying conditions of driving the motor-car engine. Investigations are in active progress to remedy the defect. In a well-designed gas engine the heat lost by imperfect combustion appears seldom to exceed 2 per cent.

(6) Combustion changes the volume of the working fluid, so that the volume which is heated is different from the volume which is compressed.

Relative Volumes of Combustible Mixtures with Oxygen and the Products of their Combustion

$$\overbrace{2\text{ vols. Ethylene } C_2H_4 + 6\text{ vols. } O}^{8\text{ vols.}} \quad \text{give} \quad \overbrace{4\text{ vols. } CO_2 + 4\text{ vols. } H_2O}^{8\text{ vols.}}$$

$$\overbrace{2\text{ vols. Butylene } C_4H_8 + 12\text{ vols. } O}^{14\text{ vols.}} \quad ,, \quad \overbrace{8\text{ vols. } CO_2 + 8\text{ vols. } H_2O}^{16\text{ vols.}}$$

$$\overbrace{2\text{ vols. Amylene } C_5H_{10} + 15\text{ vols. } O}^{17\text{ vols.}} \quad ,, \quad \overbrace{10\text{ vols. } CO_2 + 10\text{ vols. } H_2O}^{20\text{ vols.}}$$

$$\overbrace{2\text{ vols. Pentane } C_5H_{12} + 16\text{ vols. } O}^{18\text{ vols.}} \quad ,, \quad \overbrace{10\text{ vols. } CO_2 + 10\text{ vols. } H_2O}^{20\text{ vols.}}$$

$$\overbrace{2\text{ vols. Heptane } C_7H_{16} + 22\text{ vols. } O}^{24\text{ vols.}} \quad ,, \quad \overbrace{14\text{ vols. } CO_2 + 16\text{ vols. } H_2O}^{30\text{ vols.}}$$

$$\overbrace{2\text{ vols. Octane } C_8H_{18} + 25\text{ vols. } O}^{27\text{ vols.}} \quad ,, \quad \overbrace{16\text{ vols. } CO_2 + 18\text{ vols. } H_2O}^{34\text{ vols.}}$$

$$\overbrace{2\text{ vols. Alcohol } C_2H_8O + 6\text{ vols. } O}^{8\text{ vols.}} \quad ,, \quad \overbrace{4\text{ vols. } CO_2 + 6\text{ vols. } H_2O}^{10\text{ vols.}}$$

$$\overbrace{2\text{ vols. Acetylene } C_2H_2 + 5\text{ vols. } O}^{7\text{ vols.}} \quad ,, \quad \overbrace{4\text{ vols. } CO_2 + 2\text{ vols. } H_2O}^{6\text{ vols.}}$$

This change of volume is not great with coal gas and air mixtures; it does not exceed 3 per cent. contraction when using coal gas and air in ordinary dilutions. Its effect may be greater with some hydrocarbons, but its amount depends on the particular chemical sequence of the combustion of a hydrocarbon. This point is being studied by several well-known chemists. If, however, complete combustion be assumed without intermediate stages, serious volume changes occur with inflammable materials now used in internal-combustion motors, leading sometimes to contraction, sometimes to expansion.

This is seen from the table on page 330.

Thus, with an explosive mixture of butylene and oxygen, 14 volumes become 16 volumes, and with the corresponding mixture of amylene and oxygen 17 volumes become 20 volumes. In a similar mixture of pentane and oxygen, 18 volumes become 20 volumes, and with heptane and oxygen 24 volumes become 30 volumes; with octane and oxygen, 27 volumes become 34 volumes. With alcohol again, which is now coming into use in these engines, 8 volumes of explosive mixture expand to 10 volumes on complete combustion. A mixture of air and alcohol could easily give an expansion of about 6 per cent. In this respect alone, then, all change of molecular volume between the uncombined gases and the compounds formed after combustion must be carefully examined before the temperature can be estimated, either from an indicator diagram taken from an engine or from the results of an explosion in a closed vessel.

(7) The admission, transfer, and expulsion of the working fluid are not accomplished without some resistance, throttling during admission, back pressure during exhaust.

In a well-designed engine of the four-cycle type this loss should not exceed 3·5 per cent. of the IHP; but in two-cycle engines, for the reasons explained, it is necessarily greater. The best result for air resistance loss is about 6 per cent. of the indicated power, and many continental engines lose more than this.

The air resistance loss of the Diesel engine is high because of the pressure loss involved by the fuel injection by highly compressed air.

(8) Loss of heat to the cylinder walls during compression.

Losses of this kind are small. Heat so lost in four-cycle engines of moderate dimensions does not usually exceed 2 per cent. of the total heat supplied. The effect on efficiency, however, is much smaller, as heat so lost diminishes the work of compression. Heat losses generally have but small effects on efficiency in the modern gas engine.

All the points mentioned in Chapter IV. have now been considered, and data given here and in the preceding chapters from which calculations can be made for various engines operating on any cycle.

APPENDICES

APPENDIX I

DETERMINATION OF CHARGE WEIGHT IN INTERNAL-COMBUSTION ENGINES

In order to find the temperature at any given points on the indicator diagram, it is necessary to know the temperature at one point on the diagram. When this is obtained, the temperature at all other points can be readily calculated from the volumes and pressures indicated.

The most convenient temperature to determine is the suction temperature—that is, the temperature of the mixture consisting of exhaust products and fresh charge contained in the cylinder at the end of the suction-stroke. The method adopted for obtaining the suction temperature for any given speed and jacket temperature is as follows :—

If v_m be the volume of gas at normal temperature and pressure which has passed into the cylinder, and v_s the volume swept by the piston in a given time, then the charge drawn in is increased in volume in the ratio $\frac{v_s}{v_m}$; the temperature to which the incoming charge of gas and air must have been raised before the inlet valve closed can therefore be calculated.

When the charge is drawn in, there is also a quantity of products of combustion of the previous ignition left in the clearance-space, and this mixes with the fresh charge and further increases its temperature.

It will be seen that the quantity of the total fresh charge drawn in is quite independent of heat given to it by the products of combustion if the specific heats are equal, as any increase of volume of the fresh charge, due to heat given up by the products of combustion in the clearance space, is compensated for by a corresponding decrease in the volume of the products of combustion. It may therefore be considered that the ultimate suction temperature is the temperature obtained by mixing the exhaust products in the clearance space at the temperature they would reach at the end of the suction stroke if kept adiabatically separate from the incoming charge, with the fresh charge at a temperature which can be calculated from the outside temperature of gas and air when the ratio $\frac{v_s}{v_m}$ is known.

Taking Test 15, 'X' engine, from the Report of the Committee on the Standards of Efficiency of Internal Combustion Engines,[1] 8,449 cubic ft. of fluid entered per hour; the temperature of the air was 13°·8 C., and the

[1] Proc. Inst. C.E., vol. clxiii. p. 241.

barometric pressure 30 ins. The volume swept by the piston per stroke was 1·961 cubic ft., so that the piston at 165·8 revolutions per minute displaced 9,750 cubic ft. of charge per hour. The entering charge therefore increases in temperature from 286°·8 C. absolute to

$$286{\cdot}8 \times \frac{9{,}750}{8{,}449} = 332^\circ \text{ C. absolute.}$$

The temperature of the exhaust gases leaving the 'R' engine was found by Professor Callendar's thermometer to be 444° C., and this temperature is assumed to be that of the exhaust gases in the clearance space of the 'X' engine at the in end of the stroke. The volume of exhaust gases in the clearance space is 0·448 cubic ft.

The volume of fresh charge at 332° C. absolute is 1·963 cubic ft.

Therefore there is in the cylinder, at the end of the suction-stroke, a mixture of fresh charge and exhaust products, whose temperature is that given by mixing 0·448 cubic ft. at 444° C., *i.e.* 717° C. absolute, with 1·963 cubic ft. at 332° C. absolute.

Assuming that the specific weights and specific heats of the fresh charge and exhaust products are equal, then the suction temperature is:

$$\frac{1{\cdot}963 + 0{\cdot}448}{\dfrac{0{\cdot}448}{717} + \dfrac{1{\cdot}963}{332}}$$
$$= 368^\circ \text{ C. absolute ; that is, } 95^\circ \text{ C.}$$

It will be seen that the fraction in this determination which depends on the temperature of exhaust products is small, and therefore a comparatively large error in the temperature of the exhaust gases makes little difference in the result.

From the above calculation it is known that the temperature of the mixed gases in the cylinder at the end of the suction stroke is 95° C.; the pressure obtained from the light card taken on the full-load test is 30 ins., and the volume is 1·963 cubic ft. The charge-weight can, therefore, be obtained from the analysis of the exhaust gases.

To obtain from the diagram the heat discharged at exhaust it is only necessary to measure the terminal pressure. This is most conveniently done by continuing the expansion line from the point where the exhaust valve opens, to meet the ordinate at maximum volume, as shown by the dotted line to the point *e*. The terminal pressure for the 'X' engine, card *a*, is thus shown to be 53·5 lbs.

The absolute temperature at *e* is

$$\frac{53{\cdot}5}{14{\cdot}7} \times 368 = 1338^\circ \text{C. absolute.}$$

The temperature drop at exhaust to the temperature of the charge is therefore 970° C.

The charge-weight may also be calculated from the data given in Tables VII. and X. of the Report of the Committee. For instance, in Test 15 of the 'X' engine the engine was running 165·8 revolutions per minute.

$$\frac{165{\cdot}8 \times 60}{2} = 4{,}974 \text{ piston strokes per hour.}$$

Total mixture per hour = 8,449 cubic feet.

Therefore

$$\frac{8,449}{4,974} = 1\cdot698 \text{ cubic ft. of mixture per charge.}$$

The explosions per hour were 4,284, and the gas supplied was 785 cubic ft. per hour. Therefore

$$\text{Gas per explosion} = \frac{785}{4,284} = 0\cdot183 \text{ cubic ft.}$$

Hence $1\cdot698 - 0\cdot183 = 1\cdot515$ cubic ft. of air per explosion stroke.
And air supplied in mixture per hour

$$= 1\cdot515 \times 4,284 = 6,490 \text{ cubic ft.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$\text{Total air supplied per hour} = 7,664 \text{ cubic ft.} \quad . \quad . \quad (2)$$

$$\left.\begin{array}{ll}\text{But weight of dry air per hour . . } = & 583\cdot7 \text{ lbs.} \\ \text{And weight of water vapour per hour } = & 4\cdot53 \;\;\text{,,}\end{array}\right\} \text{Table VII.}$$

$$588\cdot23 \text{ lbs.}$$

∴ Weight of air and water vapour per hour for explosion strokes from (1) and (2)

$$= \frac{6,490}{7,664} \times 588\cdot23 = 498\cdot0 \text{ lbs.}$$

$$\text{Weight of 785 cubic ft. of gas} = \;\; 28\cdot9 \text{ lbs. per hour.}$$

$$526\cdot9 \text{ lbs.}$$

∴ Weight of entering charge

$$= \frac{526\cdot9}{4,284} = 0\cdot123 \text{ lb.}$$

Taking 444° C. as the temperature of the exhaust gases which mix with the entering charge,
Weight of exhaust gases

$$= 0\cdot448 \times 0\cdot0782 \times \frac{273}{717} = 0\cdot014.$$

Therefore total charge-weight

$$= 0\cdot123 + 0\cdot014 \text{ lbs.}$$
$$= 0\cdot137 \text{ lbs.}$$

The composition of the exhaust gases was as follows :—

	Cubic ft.	Lbs.
H_2O . .	0·12	0·00603
CO_2 . .	0·05	0·00619
O . . .	0·08	0·00716
N . . .	0·75	0·05895
	1·00 weight of 1 cub. ft. =	0·07833

therefore $0\cdot0783 \times 2\cdot41 = 0\cdot189$ lb. = weight of charge of 2·41 cubic ft. at 0° C. and 1 atmosphere.

The temperature of the charge was 368° C. absolute, so that the weight of the charge at that temperature calculated from the exhaust-gas analysis is

$$\frac{273}{368} \times 0\cdot189 = 0\cdot140 \text{ lb.}$$

This agrees closely with the value found above.

APPENDIX II

CALCULATION OF ADIABATIC LINES WITH VARYING SPECIFIC HEAT OF WORKING FLUID

Assuming that the apparent specific-heat values given on p. 235 are the true values of the specific heat for the temperature range, the true adiabatic line between the temperatures given can be determined. This is most conveniently done by calculating the mean values of γ for small temperature ranges, and by using these values of γ to obtain the required curve.

If C_p and C_v be the specific heats at constant pressure and constant volume respectively in foot-pounds per cubic foot at 0° C. and 14·7 lbs. per sq. in. pressure; since $C_p - C_v$ is equal to the work done by a cubic foot of the working fluid in expanding against atmospheric pressure 14·7 lbs. per sq. in., while the temperature is raised through 1° C.; and since the working fluid, assumed to be a perfect gas, expands through $\frac{1}{273}$ part of its volume for a temperature rise of 1° C.,

$$C_p - C_v = \frac{14{\cdot}7 \times 144}{273} \text{ ft.-lbs.}$$

$$\text{whence } p = \frac{C_p}{C_v} = 1 + \frac{14{\cdot}7 \times 144}{273 \times C_v}$$

$$= 1 + \frac{7{\cdot}76}{C_v} \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

In order to obtain the adiabatic it is necessary to know approximately the mean temperature on the part of the curve being constructed. It is therefore necessary, in drawing an adiabatic between any two given volumes, say, V_0 and V_1, to assume values of γ, and so obtain an approximation to the adiabatic, the temperatures on this approximate adiabatic being then taken in order to select the proper value of γ for the final curve. It is convenient, in constructing the adiabatic, say, between temperatures 1600° C. and 1000° C., to divide the volume into a number of parts, say six, and in the first instance assume that the mean temperature on the curve decreases regularly between the volumes taken. Numbering the ordinates 1, 2, 3, 4, and 5, on the assumptions made, the temperature at V_0 is 1600° C. At ordinate 1 the temperature will be 1500° C. Therefore it is required to draw between the volume V_0 and the volume at ordinate 1 the adiabatic with a specific-heat value equal to the mean value between the temperatures 1600° and 1500° C.

From equation 1 given above, and the specific-heat Table, it is seen that the value of γ for the temperature range 1600° to 1500° C. is

$$1 + \frac{7{\cdot}76}{27{\cdot}5} = 1{\cdot}2825.$$

The point where the adiabatic cuts the ordinate 1 can therefore be found from the equation

$$P_v{}^{1{\cdot}2825} = \text{constant}.$$

Thus a first approximation is obtained to the adiabatic between the volumes V_0 and V_1. From the pressure at ordinate 1 obtained in this way, the temperature on the adiabatic at 1 can be got, and, if necessary, a further approximation to the value of γ, which should be taken between the volumes considered. The specific-heat changes, however, are not sufficient to make it usually necessary to

go further than a single approximation. To get a further point on the adiabatic, the process above described is repeated for the volumes V_1, V_2, taking, as starting-point, the known point on the ordinate at volume 1, and so on. Thus the adiabatic can be found for any temperature range within the temperatures given in the Table of specific heats.

APPENDIX III

CALCULATION OF EFFICIENCY WITH VARYING SPECIFIC HEAT OF WORKING FLUID

Assuming that the apparent specific-heat values are true specific heats, the ideal efficiency of any engine can readily be calculated.

With varying specific heat, however, the efficiencies will alter as the values of maximum temperature are altered, and it is therefore necessary, in calculating

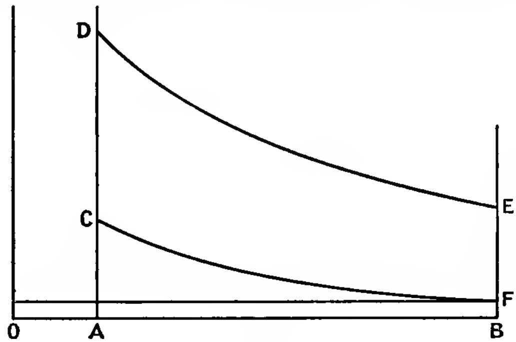

FIG. 120

efficiency, to take definite values of maximum temperature, as well as a definite ratio of clearance space to total volume of cylinder. The value of γ for the compression line may be assumed to be 1·37.

In the figure let

OA be the clearance volume,

OB, total cylinder volume,

D, point of maximum temperature,

DE, adiabatic through D,

FC, compression line.

Then it is known that the efficiency, η,

$$= \frac{\text{Heat added from C to D} - \text{Heat discharged E to F}}{\text{Heat added from C to D}}$$

$$= \frac{(T_D - T_C)\, C_{v\,(CD)} - (T_E - T_F)\, C_{v\,(EF)}}{(T_D - T_C)\, C_{v\,(CD)}}$$

$$\therefore \eta = 1 - \frac{T_E - T_F}{T_D - T_C} \frac{C_{v\,(EF)}}{C_{v\,(CD)}}$$

from which the values of η, given in the Table, can readily be obtained.

APPENDIX IV

GASEOUS EXPLOSIONS

First Report of the Committee, consisting of Sir W. H. Preece (Chairman), Mr. Dugald Clerk and Professor Bertram Hopkinson (Joint Secretaries), Professors Bone, Burstall, Callendar, Coker, Dalby, Dixon, Hele-Shaw, Smithells, and Watson, Dr. Harker, Lieut.-Colonel Holden, Dr. Petavel, and Captain Sankey, appointed for the Investigation of Gaseous Explosions, with Special Reference to Temperature.

British Association, Section G, Dublin 1908.

General Scope of the Report

To engineers the investigation of gaseous explosions is chiefly of interest because of its bearing upon the theory of the internal-combustion engine. The Committee have hitherto considered it mainly from this point of view, conceiving that a limited interpretation of their reference would be necessary if their labours were to lead to any result within a reasonable time, and that the limitation adopted should be determined by the fact that the Committee was initiated by the Engineering Section. On the other hand, the work has been by no means entirely, or even mainly, of a practical as distinct from a purely scientific character. Many questions of a kind that might properly engage the attention of the Chemical and Physical Sections have been raised and discussed, and full scope has been given to the varied skill and knowledge possessed by the different members of the Committee, among whom, in addition to engineers, there are several whose interests are mainly in the direction of pure science. The test of practical value or interest has only been applied for the purpose of selecting from among the large number of questions arising in connection with explosions those which are proper subjects for investigation by this Committee, not with the idea of limiting such investigation to the practical aspect of these questions.

Seven meetings of the Committee have been held, and they have been excellently attended. At each meeting one or more notes written by members of the Committee have been presented and have formed the basis of the discussion. The following is a list of these notes :

No. 1.	General Introduction	Dugald Clerk.
No. 2.	Dissociation of Steam and Carbonic Acid . .	Dugald Clerk.
No. 3A.	Measurements of Internal Energy of Gases up to 1400° C.	B. Hopkinson.
No. 3B.	Explosion Pressures as a Means of Determining the Energy Function of Gases	B. Hopkinson.
No. 4.	Dissociation and Specific Heat of Steam and Carbonic Acid, and Comparison of Gas-Thermometers at very High Temperatures .	J. A. Harker.
No. 5.	The Temperature of the Walls of a Gas-engine Cylinder	E. G. Coker.
No. 6.	The Deviation of Actual Gases from the Ideal State, and the Experimental Errors in the Determination of their Specific Heats . .	H. L. Callendar.

The essential feature common to the operation of all gas-engines is the conversion of a mixture of inflammable gases, by combustion or explosion, into

a mass which consists in all practical cases of a mixture of steam, carbon dioxide, nitrogen, and excess oxygen. The performance of the engine depends primarily on the change in pressure or volume, or both, resulting from this chemical transformation, and on the properties of the products of the transformation after they are formed. It depends in only a secondary degree on the nature of the chemical process and on the velocity with which it takes place. These matters, important though they must be in any investigation of explosions and in the theory of the gas engine, are not of the first importance. The foundation must be a knowledge of the properties of the gases enumerated above at the temperatures occurring in the gas engine—that is, between 1000° and 2500° C. This Report therefore consists mainly of an analysis of the present state of knowledge on this subject, together with suggestions as to the directions in which further research may be undertaken with the object of advancing it. It will be found, however, that as the mechanism and the velocity of combustion must be taken into account as disturbing factors when applying our knowledge of the gases in the theory of the gas engine, so they enter into many of the experiments on which that knowledge is based, and some discussion of them is indispensable in any criticism of these experiments.

Thermodynamic theory shows that the physical properties of a gas in chemical equilibrium are completely specified when—

(1) The relation between the pressure and volume at constant temperature is known, and

(2) The internal energy per unit volume is given as a function of the temperature and the density.

The energy of a gas per unit of mass at temperature θ is usually defined as $k\ (\theta - \theta_0)$, where θ_0 is the standard temperature from which energies are reckoned, and k the mean specific heat at constant volume between the temperatures θ_0 and θ. The second of these data is therefore equivalent to a knowledge of the specific heat in terms of the temperature and density. This form of statement is probably more familiar; but, for reasons given later on, it is in many ways less convenient than that based upon the energy function.

For those gases with which we have to deal it may be assumed that the first relation is that given by Boyle's Law. Experiment and theory alike point to the conclusion that deviation from this law only occurs when the density of the gas departs widely from its normal value, and that it is diminished by high temperature. In the gas engine the density of the gas rarely exceeds ten times that of the atmosphere, a point at which the deviation from Boyle's Law in air (at 100° C.) is only about one-half per cent.[1]

It is usual to make the further assumption that the product pv is proportional to the absolute temperature θ. A detailed examination of the grounds of this assumption forms the subject of a section of this Report. At this point it is only necessary to notice that, if it be true, then the internal energy is a function of the temperature only, and is independent of the density. If, on the other hand, the perfect gas law does not hold, then the true relation between pv and θ can be deduced from a knowledge of the internal energy, which is in that case a function both of the temperature and of the density.

The properties of the gases with which we have to deal are therefore completely defined when the energy has been tabulated as a function of the temperature and the density. So far as the present state of knowledge goes, the energy is to be expressed in terms of temperature only; but an important part of future

[1] See Witkowski, *Phil. Mag.*, vol. xli. (1896), p. 309.

investigation must deal with its dependence on the density either by direct measurement or by a determination of the relation between pv and θ at high temperatures.

The prediction of the temperature reached in combustion, which must be the starting-point of any investigation of explosions, also rests primarily upon a knowledge of the energy function. For, subject to corrections for loss of heat, incomplete combustion, and work done while combustion proceeds, the thermal energy of the mixture of steam, CO_2, &c., after combustion is equal to the chemical energy of the gases from which that mixture was formed. The latter can be accurately inferred from the composition of the combustible gases, and, the thermal energy being thus known, the temperature can be calculated from a table of the energy function. The pressure or volume changes resulting from combustion can be deduced from the temperature by the use of the p—v—θ relations, which again ultimately depend upon the form of the energy function. A table of this function at high temperatures is therefore the first datum necessary for the investigations entrusted to the Committee, and is the principal subject of this Report. Before proceeding to the discussion of this physical question, however, it is well to say something further about its bearing on practical engineering problems.

The first requisite for predicting the performance of a gas engine is to know the rise of temperature and the consequent rise of pressure produced by the explosion. The importance of this need not be insisted upon; it is not only the principal factor in the mean pressure developed, it also determines in large measure the mechanical design of the engine and the necessary strength of its parts. The part played by the energy function in the calculation of this rise of pressure has been indicated in the last paragraph. In proceeding further to analyse the indicator diagram given by the engine with the object of accounting at each point for the heat which has been put in, a knowledge of this function is again required. The heat accounted for on the diagram is the work which has been done plus the heat contained in the gas. The latter item can be calculated from the temperature if the energy function be known. The balance unaccounted for, which it is usually the object of such investigations to find—whether in the steam engine or the gas engine—is the heat which has been lost to the walls or has been suppressed owing to incomplete combustion. In fact, the internal energy of the gases at high temperatures plays much the same part in the analysis of gas-engine phenomena as does the total heat of steam in investigating the working of the steam engine.

Again, from a table of internal energy, it is possible to predict the pressure changes resulting from any series of operations such as occur in the gas engine, one item in which is an explosion subject to certain hypothetical conditions which cannot be realised in practice, though they can be indefinitely approached. An ideal diagram of this kind, corresponding to the cycle of operations which is most usual in present-day gas engines, can, for example, be constructed for any given combustible mixture on the assumption that the combustion is instantaneous and complete at the in-centre; that there is no loss of heat in compression, explosion, or expansion; and that during expansion the gases are at all times in thermal and chemical equilibrium. These conditions can never be completely realised, but can in theory be approached asymptotically by improvements in design carried on within certain defined limits—namely, that the degree of compression and the nature of the mixture are to be unaltered. For example, the heat loss may be reduced by increasing the size of the engine and altering the

nature of the cylinder walls, and the attainment of thermal and chemical equilibrium may be promoted by reducing the speed. Such an ideal cycle is, in fact, precisely analogous to the Rankine cycle of the steam engine, in that it takes account of the actual physical properties of the working substance, but leaves out of account such non-essential imperfections as heat loss to the cylinder walls. It represents an ideal which the real engine may approach indefinitely but can never attain ; and the closeness of the approach is a true measure of the perfection of the engine.

The ideal cycle which has hitherto been used in discussing the performances of gas engines is the well-known air cycle. This is based upon a special assumption as to the form of the energy function—namely, that it is a linear function of the temperature at high, as it is known to be at low, temperatures. The specific heat of the working substance is taken to be constant and equal to 19 foot-pounds per cubic foot, or 4·8 calories per gramme molecule. In the state of ignorance as to the real form of the energy function which prevailed until quite recently, this assumption was as good as any other, since it was impossible to say that the value of the energy derived from it was further from the truth than any other value which might be assigned to it. So far as was known, the differences between the indicator diagram of a real engine and the corresponding air-cycle diagram might have been wholly, or almost wholly, due to what have been called above 'non-essential imperfections'—that is, to heat loss and to incomplete combustion. In other words, there was no conclusive evidence that the air cycle was not for practical purposes a true ideal cycle in the sense defined above and equivalent to the Rankine cycle for the steam engine. Under these circumstances its extreme simplicity made it the best available standard of comparison for judging the performance of a real engine. Recent researches, however, on the properties of the gases at high temperatures have definitely shown that the assumption of constant specific heat is erroneous, and have given sufficient information about the magnitude of the error to show that it is of material importance. They have shown that the air cycle cannot be regarded as equivalent to the Rankine cycle in the steam engine, inasmuch as it does not take account of the properties of the actual working fluid, but postulates a hypothetical fluid which has no real existence. It is as though in the theory of the steam engine the total heat of the steam were to be taken as equal to its latent heat, the sensible heat of the water being neglected. This assumption would lead to a simpler formula for the ideal efficiency for the steam engine, but it would be erroneous in the same way and to about the same extent as the air-cycle formula for the gas engine.[1] The closer approximation to the real cycle which is made by taking account of the actual properties of the working fluid—in the steam engine the total heat of the steam instead of only the latent heat, in the gas-engine the true value of the energy instead of that based on the assumption of constant specific heat—though it leads to some complication of formulæ, gives compensating advantages of real practical value. It shows the engineer what are the limits to the improvements which can be effected by changes of design or increase of size, and it enables him to judge whether it is

[1] If the sensible heat of the steam can be neglected in comparison with its latent heat, the Rankine cycle reduces to the Carnot cycle, with efficiency $\frac{T_1 - T_2}{T_1}$, for no heat is then necessary to warm the water from the condenser to the boiler temperature, and the whole process becomes reversible. The efficiency of the Carnot cycle usually exceeds that of the corresponding Rankine cycle by about one-eighth part.

better that the lines of development should proceed in such directions or in the direction of radically modifying the cycle of operations.

Measurement of the Internal Energy or Specific Heats of Gases at High Temperatures

The results of most experiments on the energy of gases have been expressed in the form of tables or formulæ giving the specific heat (referred to unit mass of the gas) in terms of the temperature. It would appear preferable for most purposes to exhibit them in terms of internal energy per unit volume. That is the form most convenient for purposes of thermodynamic calculation, and it has the further advantage that it expresses the actual quantity measured. In nearly all the experiments on the specific heats of gases the increase of energy in unit volume associated with a large rise of temperature is measured; and in most the lower limit of temperature is near that of the room. The rate of change with temperature, of the energy so determined, is sometimes called the 'true' or 'instantaneous' specific heat, and sometimes 'thermal capacity.' The Committee are of opinion that a definite name should be given to this important quantity, and they suggest the name 'volumetric heat,' which if adopted should include in its significance that the measurement to which it relates is made at constant volume, and is referred to unit volume of the gas. The term 'specific heat' could then be restricted to its usual meaning, which refers to unit mass of the substance. Convenience of calculation is promoted if the unit of volume taken is that corresponding to the gramme molecule under standard conditions, which is sufficiently nearly the same for each of the gases under consideration and equal to 22·25 litres.[1] In this report internal energy and volumetric heat are expressed as calories[2] per 22·25 standard litres; and the zero of temperature from which the energy is reckoned (except where otherwise stated) is taken to be 100° C., in order that steam may be included on the same basis as the other gases. The results are conveniently exhibited as curves in which the energy is the ordinate, and the excess of the temperature over 100° C. is the abscissa. The slope of such a curve represents the volumetric heat C, and the ordinate divided by the abscissa for any temperature represents the mean volumetric heat from 100° C. to that temperature, here denoted by $\bar{C}$.

The experimental work done on this subject may be divided into three classes:

(1) Constant-pressure experiments: Regnault, Wiedemann, Witkowski, Lussana, Holborn and Austin, Holborn and Henning. The gas is heated from an external source in these experiments, and is at atmospheric pressure.

[1] The volumes of the gramme molecule for the several gases are:

H_2	22·24	CO	22·21
N_2	22·28	CO_2	22·08
O_2	22·22		

in litres at 0° C. and under a pressure of 760 mm. of mercury.

It may be noted here that 1 calorie per gramme molecule is equivalent to 3·95 foot-pounds per cubic foot.

[2] There is some difference in the energy value of the calorie according to the temperature at which it is measured. The difference between the maximum and minimum value over the range 0° to 100° C. amounts to about 1 per cent. This is of no importance for the purposes of this Report, except in one or two places; but where it is necessary to be so precise the calorie at 15° C.—namely, the quantity of heat required to warm 1 gramme of water from $14\frac{1}{2}$° C. to $15\frac{1}{2}$° C.—is meant.

(2) Experiments in which both volume and pressure are varied, the gas being heated by compression. The recent experiments of Clerk and the determinations of the velocity of sound in hot gas by Dixon and others belong to this class.

(3) Constant-volume experiment. To this category belong the explosion experiments of Mallard and Le Chatelier, Clerk, Langen, Petavel, Hopkinson, and others, and Joly's determinations with the steam calorimeter. In the explosion experiments the gas is heated by internal combustion.

(1) *Constant-pressure Experiments*

The constant-pressure experiments have been carried to a temperature of about 1400° C. The gas under atmospheric pressure flows steadily through a heater and then through a calorimeter, where it is cooled. The temperature just before entering and just after leaving the calorimeter and the quantity of heat evolved per gramme molecule of the gas are measured. This quantity of heat less the work done in the contraction, which is 1·98 times the fall of temperature, is the change of internal energy corresponding to that fall.

Regnault applied the method to air, H_2, CO, CO_2, and other gases over the range 0–200° C.

Wiedemann [1] repeated Regnault's experiments with some modifications of the apparatus. On account of the small range of temperature these experiments must be regarded as only giving the slope of the internal energy curve at the origin ; but as they give this with an accuracy at least equal to that with which the ordinate is known at higher temperatures, they are of considerable importance in constructing the curve. The following table shows the values of the mean volumetric heat $\bar{C}$. over the range 0°–100 C., found by these two observers for air, H, and CO. Witkowski's value for air, by the same method, is in exact agreement with Regnault's : [2]

	H_2	CO	Air
Wiedemann	4·84	4·81	4·90
Regnault	4·84	4·88	4·86

These results give a good idea of the accuracy attained in these experiments. In both sets the different observations ranged about 1½ per cent. above and below the mean in each determination. Later work shows that the value of $\bar{C}$ for air is probably about 1 per cent. greater over the range 0 to 200 than over the range 0 to 100. Regnault was unable to detect this difference, though he looked for it.

The volumetric heat of air has also been determined by Joly by means of the steam calorimeter. He found the specific heat of air at constant volume for the range 10° to 100° C. and at a pressure of about twenty atmospheres to be 0·172. There were distinct signs of an increase of specific heat with density, and, assuming this to follow the linear law given by Joly, the specific heat at normal density would be 0·1715, equivalent to 4·93 calories per gramme molecule. Professor Callendar points out, however, that this is based upon Regnault's number for the latent heat of steam, which is of doubtful accuracy, and that more probably Joly's determination when reduced to the 15° calorie should be 0·1732, or 4·98 calories per gramme molecule. According to some unpublished experiments by a constant-pressure method, which have been made by Mr. Swann in Professor Callendar's laboratory, and in which it is believed that some sources of systematic error inherent in the earlier experiments of this type have been avoided, the volumetric heat of air is 5·0 These results are distinctly higher than

[1] *Annalen der Physik*, 1876, vol. clvii. [2] *Phil. Mag.* vol. xlii. (1896), p. 5.

those obtained by Wiedemann and Regnault, but the difference is of no importance for the present purpose except as an indication of the possibility of systematic errors in their method of experiment which may become important when it is applied to higher temperatures. It may be taken as fairly certain that the volumetric heat of air at 100° C. is within 2 per cent. of 4·9.

In the case of CO_2 the results obtained by Wiedemann and Regnault were :

		W	R
Increase of internal energy	0 to 100° C.	710	680
,, ,, ,, ,,	0 to 200	1510	1490
,, ,, ,, ,,	100 to 200	800	810

The first two rows of figures represent practically the quantities actually measured in these experiments.[1] The third is obtained by difference from the first two, and is therefore affected with a greater probable error than either. The result of the two sets of experiments may be summed up by saying that the volumetric heat of CO_2 at 100° C. taken as equal to the mean volumetric heat between 0° and 200° C. is between 7·45 and 7·55, and that its rate of increase with temperature is between 0·009 and 0·013, or roughly one six-hundredth part per ° C. The specific heat of steam at constant (atmospheric) pressure in the neighbourhood of 100° C., according to Regnault, is 0·48, equivalent to 6·64 volumetric heat, and subsequent observers have shown that this value is at least as accurate as Regnault's value of the specific heat of air.

From Joly's experiments with the steam calorimeter, when corrected according to Callendar for the error in Regnault's value of the latent heat of steam, the specific heat of CO_2 between 10° and 100° C. and at a pressure of 12 atmospheres is 0·172 and it increases by about 0·25 per cent. per atmosphere. Assuming this law of increase to hold between one atmosphere and 12 atmospheres, the mean specific heat at normal density for the range 10° to 100° should be 0·1666, and the volumetric heat should be 7·3, which is again decidedly greater than the values obtained by Wiedemann and Regnault. According to a recent determination by Swann, the result of which has been communicated to the Committee by Professor Callendar, the volumetric heat of CO_2 at 100° is 7·76—again materially higher than Regnault and Wiedemann (7·5).

Holborn in conjunction with Austin carried the constant-pressure determinations for air and CO_2 up to 800° C.[2] The gas was heated electrically and the temperature was measured with a thermo-couple. Similar measurements on steam were made by Holborn and Henning, who subsequently carried the determinations for the three gases up to 1400° C.[3]

Holborn and Henning express the results of all these experiments in algebraical formulæ representing the mean specific heats of CO_2, air, and steam respectively over the range $0–\theta$ in the case of the first two gases, and $100–\theta$ in the case of steam. From these formulæ the full-lined curves in fig. 121, exhibiting the internal energy, have been constructed. The actual observations are also shown in the same figure. Each of these observations represents the mean

[1] The lower limit of temperature in Regnault's measurements was 10°, the upper limits were 100° and 210° respectively. In Wiedemann's work the lower limit was 25° and the upper 100° and 200° respectively. The results have been reduced to the forms here given on the supposition that the volumetric heat between 0° and 25° is 6·6, and at 200° 7·4. For the small additional range of temperature required these figures are certainly as accurate as the experiments.

[2] *Wiss. Abhandlungen der Phys. Techn. Reichsanstalt*, 1905.

[3] *Ann. d. Phys.*, 23, 1907, p. 809.

of a large number of experiments, in some cases as many as thirty. The results of the individual experiments in such a group ranged about 2 or 3 per cent. above and below the mean. These casual errors would no doubt cancel out to a great extent in taking the mean, which, apart from systematic errors inherent in the method of experiment, is probably correct within about 2 per cent.

This degree of accuracy is not sufficient to enable any deduction to be made as to the manner of variation of the volumetric heat beyond a rough estimate of its average rate of increase over the whole range of experiment. The following

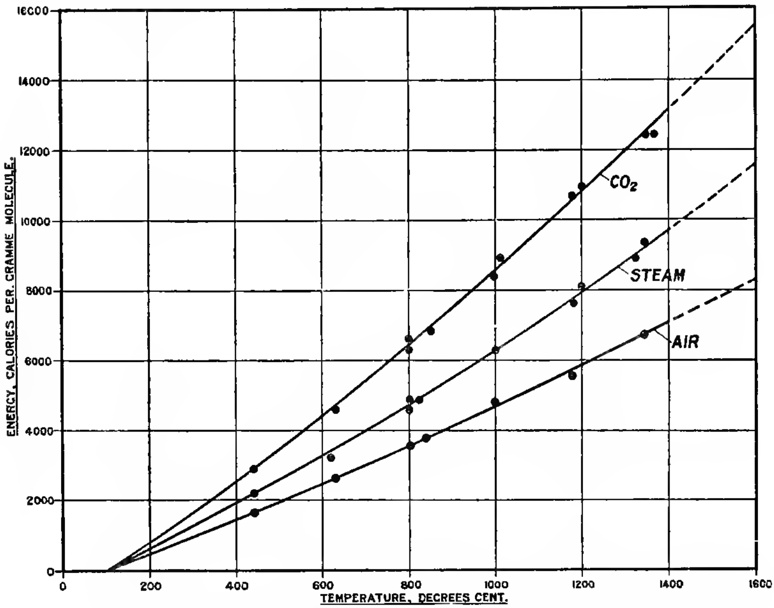

FIG. 121.—Curves showing internal energy at different temperatures from Holborn and Henning's experiments

are the values of the volumetric heats of air, steam, and CO_2 at 100°, 600°, and 1100° respectively :—

—	100°		600°		1100°		Increase 100–1100°
	C	γ	C	γ	C	γ	—
Air	4·9	1·404	5·2	1·38	5·75	1·345	0·9
Steam	6·6	1·30	6·85	1·29	8·5	1·24	1·9
CO_2	7·5	1·26	9·95	1·20	11·1	1·18	3·6

The corresponding values of γ are also shown : $\gamma = 1 + \frac{1·98}{C}$.

The values at 100° are derived from the experiments of Wiedemann and Regnault. Those at 600° and 1100° are based on the specific-heat values given by Holborn and Henning; in other words, they are obtained by drawing tangents to the curves, fig. 121. The error at these higher temperatures may be double that of the internal energy, or, say, 4 per cent. The figures show that the volumetric heat of air increases by about 0·0009, that of steam by 0·0033, and that of CO_2 by 0·0036 per degree Centigrade over the range 100°–1100° C. There is no evidence that the rate of increase is other than constant in the case of air; but there can be no doubt that the average rate of increase between 100° and 1100° in CO_2 is less than half the rate of increase between 0° and 200°, as determined by Wiedemann and Regnault. There is also distinct evidence in these and other experiments that the rate of increase of the specific heat of steam becomes greater as the temperature rises.

(2) *Clerk's Experiments* [1]

These cover about the same range of temperature as Holborn and Henning The gas used was the products of an explosion in a gas engine, and therefore

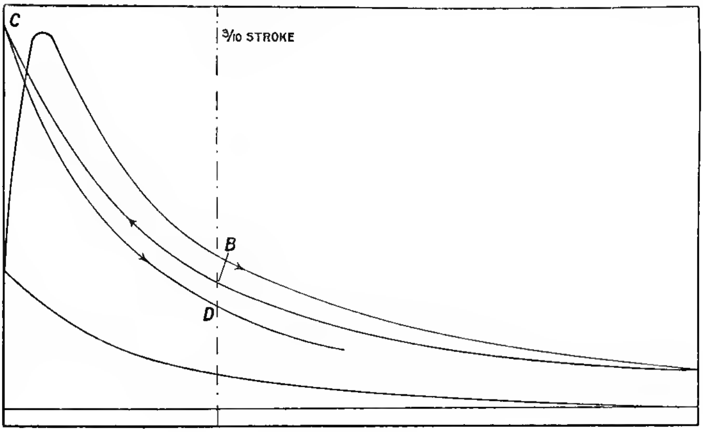

FIG. 122

consisted of a mixture of CO_2, steam, and air. It was first expanded in the ordinary course after the explosion, and was then heated by compression on the next in-stroke of the engine, the valves being kept closed for this purpose. On the next out-stroke the gas was again expanded, then compressed again, and so on, the valves remaining closed and the engine running on its own momentum. An indicator diagram was taken of the whole operation. The change of internal energy in any portion of a compression stroke (*e.g.*, B C in fig. 122) is equal to the work done less the heat lost to the cylinder walls; in an expansion stroke (C D) it is the work done plus the heat lost. The work can be obtained from the indicator diagram with an accuracy which is only limited by the indicating appliances. The change of temperature can also be calculated from the indicator diagram subject to a knowledge of the temperature at one point. Errors in the latter, however, do not greatly affect the results found for internal energy or volumetric

[1] *Proc. Roy. Soc.* A, vol. lxxvii.

heat, because the figure for the quantity of gas present is affected by these errors in such a way as to cancel out the error in temperature interval.

The loss of heat comes in as a correction on the work done and was estimated by a comparison of the compression line and the immediately following expansion line (B C and C D, fig. 122). The calculation is based on the assumption that the total heat loss from the hot gases during any given portion of a stroke is the same in expansion and compression if the mean temperature be the same.

In the first compression the temperature of the gas rose to about 1100° C. (at the point C, fig. 122). During the first three-tenths of the following expansion stroke (C D), the temperature fell to about 700° C. The work done in this part of the expansion was measured and the heat loss determined as above was added. Thus the change of internal energy corresponding to the temperature change 1100°–700° is obtained. The average volumetric heat over this range is within the errors of experiment equal to the volumetric heat at the mean temperature of 900° C., which accordingly is by this method determined direct instead of by difference, as is necessarily the case when (as in Holborn and Henning's experiments) the whole internal energy change associated with complete cooling of the gas is measured.

In view of the great difference in the method of experiment a comparison of Clerk's results with those of Messrs. Holborn and Henning is of great interest. Clerk's measurements extended to 1450° C., but those above 1200° C. were based on the first expansion line after the explosion when the method for getting heat loss would be of doubtful application, and when, moreover, combustion may have been incomplete. It will be better, therefore, to confine the comparison to temperatures of 1200° and below. The following table exhibits the internal energies of the mixed gas with which Clerk experimented calculated from Holborn and Henning's figures, together with the energy calculated from Clerk's values for the mean volumetric heat. The energies are, as usual, reckoned from 100° C.; and the energies of an ideal gas with a constant volumetric heat of 4·9 are added for comparison.

Temperature	Holborn and Henning	Clerk	Ideal Gas
400	1580	1720	1470
800	3840	4300	3430
1200	6285	7040	5390

It will be seen that Clerk's results are throughout about 10 per cent. higher than the others. The difference between the energy of the real and of the ideal gas, the discovery of which is the true object of these experiments, is about twice as great in the one case as in the other. It does not seem possible to account for so large a discrepancy by ordinary experimental errors. It must be due either to some systematic error inherent in the method of experiment in one or both cases, or to a difference in the conditions of experiment giving rise to a real difference of internal energy.

Professor Callendar has favoured the Committee with a note dealing with the constant-pressure experiments. He is of opinion that the results obtained by Regnault's method are too low, and that at the higher temperatures reached by Holborn and Henning the error may possibly amount to as much as 10 per cent. In all these experiments there is a considerable flow of heat from the heater to the calorimeter. This, of course, has to be deducted from the heat registered in the

calorimeter in order to find that which has been given up by the hot gas. All the experimenters by this method have determined the amount of this correction by observations when the gas was not flowing, or have compensated it under the same conditions by radiation from the calorimeter, making, in either case, the assumption that the amount of heat conducted is the same whether the gas be flowing or not. Professor Callendar is of opinion that this heat-flow is, in fact, much less when the gas is flowing. He considers that even the values obtained by Regnault may be as much as 2 or 3 per cent. too low, and he supports this contention by reference to the work of other experimenters (some of which has been alluded to above) and by theoretical considerations. As this type of error is likely to increase greatly with rise of temperature a systematic error of even 2 per cent. in Regnault's results, if established, would give reason to suspect that the experiments at high temperatures may be subject to errors of real importance for the present purpose.

If there be systematic error in Mr. Clerk's work it seems most likely that it lies in the estimate of heat-loss. The total heat-loss in the first partial compression and expansion line in the diagram (B C D, fig. 122) is estimated from the fall of temperature and from the net work done (area B C D) in the double operation, and amounts to, roughly, half the work done in expansion. This loss has to be divided between compression and expansion, and Mr. Clerk divides it on the assumption that if the mean temperature in compression and expansion were the same the heat loss would also be the same. The mean temperature in expansion is, in fact, rather less than in compression, and the heat loss calculated in this way is correspondingly smaller, but the difference on this account is not very great, and the result is, roughly speaking, that the loss is equally divided between the two operations. Thus the correction to be added to the work done in expansion in order to get the total loss of energy of the gas is about 25 per cent. of the work, or 20 per cent. of the energy change.

Professor Hopkinson has dealt with this point in a note which he communicated to the Committee, and he is of opinion that, relative to the mean temperature, the heat loss is really much greater in compression than it is in expansion. He supports this view by reference to some experiments which he has made on the compression and expansion of a charge of cold air in a gas engine which was motored round with the gas cut off. The specific heat of air being known, the loss of heat in any part of the compression or expansion stroke can in this case be independently estimated from the diagram. He found that while in the latter half of the compression stroke the heat lost to the walls amounted to a considerable fraction of the work done, some part of this loss was actually restored to the gas during the first half of the succeeding expansion, and this notwithstanding the high temperature of the air, which in expansion, as in compression, was much above that of the walls. An estimate of the thermal capacity could, of course, be obtained from this diagram by the application of Mr. Clerk's method, and it would lead to a result considerably in excess of the truth. Mr. Clerk has himself tried this same experiment of compressing and expanding air, and he also has found that the resulting value of the specific heat of air is too high and that the air takes in heat during expansion. Professor Hopkinson thinks it possible that the heat lost during the partial compression line in Clerk's diagram may be more than twice as great as the loss during expansion. If this were so, the correction for heat loss in expansion would be less than 16 per cent. of the work done instead of 25 per cent., and the true change of energy would be less than that calculated on the assumption of equal heat loss in compression and expansion by 7 per cent. or more.

It will be seen that the errors believed to affect each method of experiment are in such a direction as to account for the divergence of the results ; and it is quite probable that when these errors are completely allowed for, the discrepancy will largely disappear. Meanwhile the internal energy of the products of combustion in the gas engine at 1200° C., if taken as the mean of Clerk's and of Holborn and Henning's results, must be regarded as subject to a possible error of about 5 per cent.

Under these circumstances it does not seem necessary to discuss the possibility that there may be a real difference between the energy values obtained by the two methods due to the different conditions of experiment. It may be pointed out, however, that Clerk's gas was at the maximum temperature from fifteen to twenty times as dense as Holborn and Henning's. This difference in the condition of the gas is such that a comparison of the results obtained by the two methods, when freed from experimental errors, will be of great interest and importance.

(3) *Explosion Experiments*

If a combustible mixture of gases be fired in a closed vessel impervious to heat, and if sufficient time elapse to allow of the attainment of complete thermal and chemical equilibrium, the internal energy of the products of combustion after the explosion will be equal to the chemical energy before explosion. The latter is capable of accurate measurement. The temperature reached after explosion can be inferred from the pressure, assuming the gaseous laws to hold. The pressure can also be measured without difficulty and with considerable accuracy.

In the study of explosion pressures we have therefore a very convenient and simple means of getting the internal energy function at high temperatures provided that it is possible to make the necessary corrections for deducing from the pressures observed in a real explosion the pressure reached in an explosion under the ideal circumstances postulated above. Moreover, the gaseous laws on which the temperature estimations are based can themselves be checked, and if necessary corrected, by comparison of the pressures reached by mixtures of the same composition but of different densities. Thus explosion experiments are capable of furnishing a complete account of all the thermal properties of gases at the temperatures reached by combustion, subject always to the possibility of making the corrections referred to above. The difficulty of finding these corrections is, however, very great, and in consequence of the uncertainty which prevails even as to their order of magnitude, the large amount of work which has been done on explosion pressures gives but little definite information as to the specific heats of gases. Nevertheless, it is to the study of explosion pressures that we owe such knowledge as we possess of the energy function at the temperatures which prevail in the gas engine, and it is to work on these lines that we must look in large measure for extension of our knowledge. A full discussion of what has been done already must therefore form an important part of this Report.

Let H be the calorific value of the mixture before combustion, let h be the heat lost at some point A on the record (taken on a revolving drum) connecting the pressure and the time (fig. 123). The energy in the gas is then $H - h$. The gas at this point is, however, certainly not in thermal equilibrium, and is probably neither in chemical equilibrium nor at rest. If therefore the loss of heat were suddenly arrested at A, the pressure would change owing to the more or less gradual attainment of equilibrium in all three respects. The equilibrium value of the pressure would be reached asymptotically, as shown by the dotted line.

When equilibrium has been attained the energy of the gas is all thermal and equal to H − h, and the temperature can be calculated in the ordinary way from the pressure. The problem, therefore, is first to find or estimate the heat loss h which has occurred at some point on the explosion record, and then to find or estimate by how much the equilibrium value of the pressure, if there were no further heat loss, would differ from that shown on the record.

This change of pressure, marked p on the diagram, is due partly to the combustion of the gas remaining unburnt at A and partly to the equalisation of temperature by convection. It may also be due to some extent to the damping down of the motion of the gas set up by the explosion.

The sooner the point A is taken, the less will be the loss of heat; but the greater, on the other hand, will be the departure from equilibrium conditions. The principal workers in this field, Mallard and Le Chatelier and Langen, assumed that the latter might be neglected if the point A were taken at the point of inflexion on the falling curve, and they estimated the loss of heat by prolonging

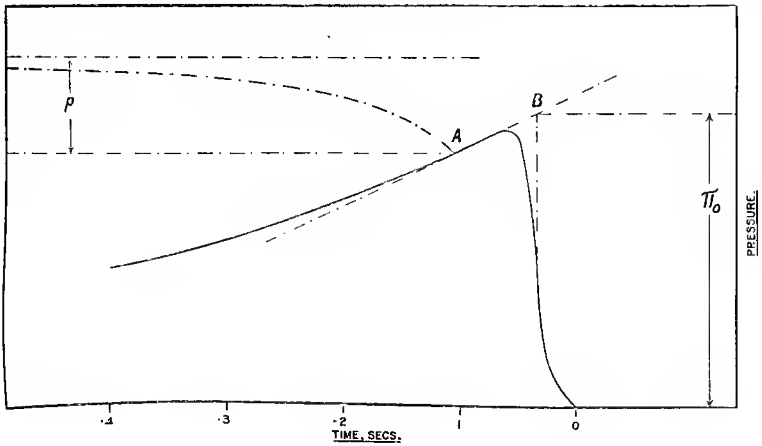

FIG. 123

this curve backwards, as shown. They assumed, in fact, that the pressure given by the point B was that which would ultimately have been reached had the explosion taken place in a vessel with walls impervious to heat.

It is difficult to justify this procedure on *à priori* grounds; the only satisfactory justification is to show, by independent evidence, that it leads to correct results. The main object of this section of the Report is to examine such evidence as there is of this kind, to point out the defects in it, and to suggest experimental methods by which they could perhaps be remedied.

In so far as the heat loss and the departure from equilibrium are dependent on surface phenomena, a definite estimate of their amount can be obtained by a comparison of explosions of the same mixture in vessels of different sizes.

Many years ago Berthelot tried this experiment, firing hydrogen and oxygen, in explosive proportions, in vessels of 300 c.c. and 4000 c.c. respectively. It is stated that the pressure reached was very nearly the same, which would show that such part of the cooling and other corrections as depends on the surface of the vessel is small in the case of this mixture.

Materials for a more accurate comparison are to be found in the extensive researches of Mallard and Le Chatelier, and of Langen. The French experimenters worked with a cylindrical vessel 17 cm. × 17 cm., whereas Langen used a sphere 40 cm. diameter. The ratio $\frac{\text{surface}}{\text{volume}}$ was 2·3 times as great in the first as in the second case.

The following table shows the results obtained in two instances, in each of which the composition of the mixture was practically identical in the two sets of experiments :—

Mixture	Observer	Πo	Cooling correction
2 vols. air, 1 vol. (H_2+O)	Mallard and Le Chatelier,	7·40	14 per cent.
	Langen	7·50	8 ,,
2 vols. air, 1 vol. $(CO+O)$	Mallard and Le Chatelier,	7·50	$7\frac{1}{2}$,,
	Langen	7·50	8 ,,

Π*o* is the pressure reached in the explosion in atmospheres after correcting for cooling in the manner described above, when the initial temperature is 0° C.

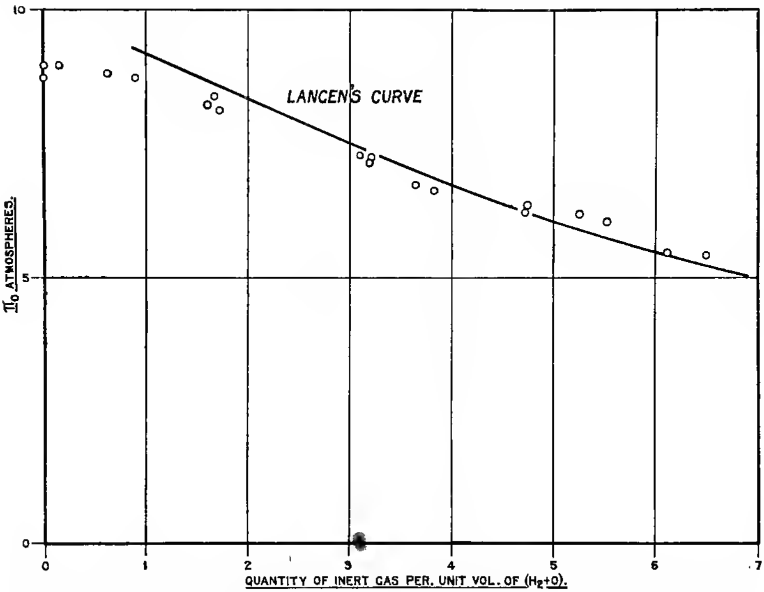

FIG. 124.—Comparison of maximum pressures corrected for cooling from Langen and Mallard and Le Chatelier's experiments. H_2O+O+ Inert Gas

The cooling correction, or excess of the pressure at B (fig. 123) over that at A, is shown in the last column. Figs. 124 and 125, which are taken from Langen's paper, show a comparison between the curve adopted by Langen, as representing the results of his experiments, and Mallard and Le Chatelier's observations.

On the whole, the agreement between the two sets of experiments is very fair,

and the deviations are not such as to suggest that any very great error has been made in estimating such part of the corrections for heat loss or for unburnt gas as depend on the surface of the vessel. If, for example, Langen were, on the average, 4 per cent. out from this cause, Mallard and Le Chatelier would be 9 per cent. out, and would differ by 5 per cent. from Langen. Differences of that amount do occur, but they do not seem to be systematic. Further experiment of the same kind on vessels with a greater difference of size but of similar geometrical form is, however, desirable.

The question remains, how far the corrections really are surface corrections. This appears to the Committee to be the most important question of a general character awaiting solution in connection with gaseous explosions when regarded as a means of investigating the properties of gases at higher temperatures. It

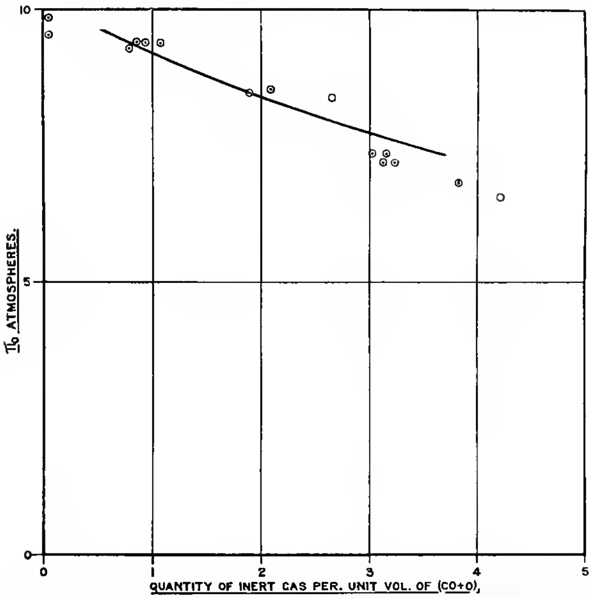

FIG. 125.—Comparison of maximum pressures corrected for cooling from Langen and Mallard and Le Chatelier's experiments. CO + O + Inert Gas.

will be convenient to discuss each of the corrections enumerated above with special regard to this question.

Loss of Heat.—That much of the heat loss goes on by direct conduction to the walls, and is, therefore, a surface phenomenon, is obvious. But there is reason to believe that the loss by radiation, which certainly exists in any flame, is practically important.

(*a*) Measurements of the temperature reached in an explosion by means of a platinum thermometer, under circumstances which render very improbable any loss of heat by conduction from the gas whose temperature is measured, show that that temperature is considerably lower than is to be expected from the heat of combustion of the gases and the specific heat of the products.

Professor Callendar pointed out, in the discussion of these experiments, that there was probably a good deal of radiation, and stated that he had found that an ordinary Bunsen flame might radiate up to 15 per cent. of its heat.[1]

(*b*) Recent experiments, in which the loss of heat during an explosion was directly measured by finding the rise of temperature of the walls, showed that in a certain coal-gas explosion it amounted to about 12 per cent. of the whole heat at the moment of maximum pressure. Estimated by Mallard and Le Chatelier's extrapolation method, the loss was at most 5 per cent.[2]

The prevailing opinion seems to be that most simple gases cannot be made to radiate by direct heating. If this be so, the radiation must take place in the act of combustion. It seems very probable that when, say, hydrogen and oxygen combine a certain part of the energy of combination passes into the form of internal vibrations of the steam molecule, and that a large proportion, if not all, of this part is ultimately radiated away. If this be the case a definite proportion of the heat produced in combustion is always lost, and a comparison of explosions in vessels of different sizes would not reveal this loss.

Thermal Equilibrium.—When an explosive mixture of gases is ignited in a closed vessel the effect of the change of pressure during the progress of the flame from the point or points of ignition is to raise the temperature round about those points much above the mean temperature, and, on the other hand, the temperature attained at those places which are last reached by the flame, and where the gas is compressed before instead of after ignition, is much below the mean. Even in a vessel whose walls are inpervious to heat the difference of temperature between the points first and last inflamed might amount to 700° C. at the moment of maximum pressure.[3] In a real explosion the cooling effect of the walls causes the temperature to range from perhaps 300° or more above the mean (as shown by the pressure) right down to the wall temperature at points close to the metal. The existence of large temperature differences in the gas close to the walls of an engine cylinder was first experimentally demonstrated by Professor Burstall with the aid of platinum thermometers.

If the volumetric heat of the gas were constant the equalisation of these temperature differences by convection and conduction, could it take place without loss of heat, would cause no change of pressure. The volumetric heat is, however, not constant, but may quite possibly be 50 per cent. greater in the hottest than in the coldest part of the mass. The attainment of thermal equilibrium must, in fact, cause a change of pressure, and would contribute to the correction which has been designated p (see fig. 123). The amount of the change might be the subject of rough calculation, taking an assumed distribution of temperature and assuming values for the volumetric heat. Such a calculation in the present state of knowledge would only be of value as showing the possible order of magnitude of the quantity sought, and the assumptions made could therefore be of a character to make the calculation fairly simple. More accurate knowledge both of temperature distribution and of thermal capacity will enable greater accuracy to be attained in the estimation of this correction, which will be of such a kind that a method of successive approximation can be pursued, the revised values of thermal capacity resulting from its application being applied to a more accurate calculation of the correction if necessary.

The temperature variation set up by the cooling action of the walls is a surface phenomenon, and as such the correction which it necessitates can probably

1 *Proc. R.S.*, A, vol. lxxvii. p. 400. 2 *Ibid.* A, vol. lxxix. p. 147.
3 *Ibid.* A, vol. lxxvii. p. 389.

be determined and eliminated by experiments with vessels of different sizes. The variation caused by the change of pressure during the period of inflammation is not of this character ; and the necessity for a large correction on this account is quite consistent with the observations of Berthelot, or of Mallard and Le Chatelier and of Langen. In these experiments the maximum pressure reached in the explosion was measured, and at the time of maximum pressure very large differences of temperature are known to exist at a distance from and quite independent of the walls.

Soon after maximum pressure, however, the temperatures at points remote from the walls are equalised to a large extent by convection currents. There then remains only the layer of gas near the walls to be considered in this connection. If, therefore, the measurements be postponed until a long enough time has elapsed to admit of this internal equalisation, the correction becomes of the surface kind, and can be dealt with by the method appropriate to corrections of that type. But in that case the heat lost will be too large a quantity to admit of rough estimation ; it must be directly measured.

Chemical Equilibrium.—The view that chemical equilibrium is not attained until some time after the moment of maximum pressure was first put forward by Clerk in 1885, who then expressed the opinion that the greater part of the so-called 'suppression of heat' in explosions was to be ascribed to this cause. On the other hand, Continental writers have almost completely ignored it. For example, Langen makes practically no reference to this in his paper. It can hardly be doubted, however, that in many explosions, especially of weak mixtures, a considerable amount of the energy is in the chemical form at the moment of maximum pressure. On the other hand, it seems probable to the Committee that the amount of unburnt gas at this moment in such experiments as those of Langen was not such as very greatly to affect the results. This belief is based on the supposition that the incomplete combustion is due to the cooling action of the walls. It seems probable that very shortly after the attainment of maximum pressure, that is, within a time small compared with that required to reach maximum pressure, the transformation of the chemical energy into thermal form is everywhere complete except in a thin surface layer where this transformation is retarded by the cooling action of the walls.

If this view be accepted, the correction of the results for incomplete combustion is of the nature of a surface correction, and can be determined by comparing the pressures reached by the same mixture when exploded in vessels of different sizes.

In the discussion of this important matter the Committee have derived great assistance from the experience of Professors Dixon and Bone, who have made a special study of the velocity of chemical action in gases. These gentlemen are of opinion that though such action may be of great complexity, involving in many cases several successive molecular operations, yet, if it is not retarded by the presence of cold foreign bodies, it will generally be completed within a period which, for the purposes of gas-engine theory, may be regarded as negligibly small. In the simple case of the explosion of hydrogen and oxygen they consider that the complete transformation of the mixed gases into steam at any given point is complete within a time measured by the interval between molecular collisions. When the action is more complicated, as in the explosion of carbon monoxide and oxygen in the presence of water, or in the combustion of hydrocarbons, the period will be larger, but will still be measured by thousandths of a second.

Some direct evidence that incomplete combustion in an explosion is mainly,

if not entirely, a surface phenomenon is to be found in Hopkinson's measurements of the temperature at points within a large explosion vessel by means of a platinum thermometer. A photographic record of the resistance of a fine platinum wire immersed in the gas showed that when the flame reached it the temperature rose in less than $\frac{1}{40}$th of a second from 20° C., which was the temperature of the unburnt gas, to about 1250° C., which was that of the burnt gas, and that it remained at the latter figure quite steadily except in so far as the increase of pressure in the vessel caused it to rise. In other words, there was no increase of thermal energy except that due to work done upon the gas from outside.[1] The mixture was one part of coal gas to nine parts of air—a slow burning mixture—and the time taken to reach maximum pressure was about a quarter of a second or at least ten times that required for combination of the gases at any one point. It is true that the vessel was of rather large size—about 6 cubic feet capacity—but, on the other hand, owing to the fact that the platinum wire extended over about 1 cm., so that the flame took an appreciable time completely to envelop it, it is probable that the period of $\frac{1}{40}$th of a second, given above, is a superior limit which greatly exceeds the actual time taken to effect the combination at any one point.

On the other hand, it cannot be doubted that combustion must be greatly retarded in the neighbourhood of the cold metal walls ; and there is nothing to show that this surface retardation is not sufficient to account for all the phenomena of delayed combustion. A simple calculation based upon the rate of flow of heat per square foot into the metal of a gas-engine cylinder (which is roughly known from measurements of the heat carried away by the jacket water) shows that the mean temperature of the exposed surface at points separated by an inch from the cooling water cannot exceed quite a moderate value. Probably about 200° C. is a superior limit for the cylinder liner. Similar calculation of a still rougher kind, but still sufficiently accurate to give the order of magnitude of the quantity sought, shows that the fluctuation above and below the mean in the course of a cycle is very unlikely to exceed 20° C. The latter conclusion has been confirmed by some experiments made by Professor Coker with a preliminary account of which he has favoured the Committee. Measuring the cyclical variation of temperature of the inner surface of a 12-h.p. gas-engine cylinder by methods similar to those adopted by Professors Callendar and Nicholson in their well-known work on the steam engine, he found that the maximum was only 7° F. in excess of the mean. The direct measurements by Professor Hopkinson of the temperature of the walls of an explosion vessel lined with copper strip also lead to the conclusion that it is quite moderate. This cold metal must obviously profoundly affect the combustion in its neighbourhood. In a layer of gas of appreciable thickness the combustion will be of a smouldering character, depending upon the velocity with which the unburnt gas in contact with the walls can diffuse into the hotter regions at a distance from them, and so be brought to the ignition temperature. This layer being cold and highly compressed might account for a considerable fraction of the heat, though its actual thickness may be only a few tenths of a millimetre. It would appear probable that the continued burning which undoubtedly goes on after the time of maximum pressure in many explosions, and probably also occurs during the first portion at least of the expansion stroke of a gas engine, is mainly of this character.[2]

[1] *Proc. R.S.*, A, vol. lxxvii. p. 387.

[2] Professor Bone is doubtful whether 'smouldering combustion' plays so considerable a part in gaseous explosions as is here suggested.

Motion of the Gas.—In many explosions intense vibratory motions of the gas are set up. The effect of these sometimes appears with a quick-period indicator as a rapid variation of pressure. It is a question of some importance how these motions affect the *mean* pressure shown by a gauge. The damping down of the motion which occurs in consequence of viscosity of course only means that the motion becomes distributed among the molecules in a random way, instead of following a definite arrangement. The total kinetic energy remains the same. But it is not certain that the mean effect on a pressure gauge of the molecular impacts will be the same. This is a question which might be considered by someone to whom the methods of the kinetic theory of gases are familiar. It is of course not a surface phenomenon.

Results of Observations.—The temperatures reached in these explosion experiments range from about 1300° up to 3000° C. Temperatures of below 1500° are, however, obtained by the use of weak mixtures, involving slow burning and large cooling corrections, and but little reliance can be placed on the results. Langen made very few observations on mixtures giving a lower temperature than 1500°, and takes that as the lower limit of the range of temperature to which his observations apply. The extreme upper limit of the constant pressure experiments is 1400°. The temperature of 3000° C. is about that reached in the explosion of hydrogen and oxygen in their combining proportions. This is much above the mean temperature ordinarily reached in the gas engine, the upper limit of which may be put at about 2000° C., though it is probable that 2500° or more is occasionally reached locally. Langen, however, places the upper limit of the application of his formulæ at 1700° C., on the ground that there is dissociation of the CO_2 at higher temperatures than that. There does not seem to be much reason for this limitation, for the effects of dissociation (provided that equilibrium is attained) are indistinguishable from those of increasing specific heat, and should be included in the change of energy. Dissociation may give rise to errors in the temperature measurement, but there is reason to suppose that the dissociation which occurs in the CO_2 in steam at a temperature of 2000° C. is too small to cause any material change of volume, though it may mean considerable absorption of heat.

The formulæ given by Langen as representing the results of his observations are as follows:

$$\text{Air } \bar{C} = 4{\cdot}8 + 0{\cdot}0006\,t$$
$$CO_2\ \bar{C} = 6{\cdot}7 + 0{\cdot}00260\,t$$
$$H_2O\ \bar{C} = 5{\cdot}9 + 0{\cdot}00215\,t$$

where $\bar{C}$ is the mean thermal capacity over the range 0 to t° C. The explosion pressures predicted by the use of these formulæ agree well with the observed pressures except in the case of mixtures of CO and air, where they are a good deal too high. In the other cases the maximum deviation is about 4 per cent.

Mallard and Le Chatelier represent their results by formulæ which differ greatly from the above in the case of CO_2 and H_2O, though the formula for air is the same. This discrepancy must be due in some way to the method of reduction adopted, for, as already pointed out, the explosion pressures reached with mixtures of the same composition are very nearly the same.

Taking Langen's values, the following table exhibits the energy of the various simple gases, and of the mixture on which Clerk experimented, at 1600° and 2000° respectively. The energy of the same gases at 800° and at 1200° based on Holborn and Henning's and on Clerk's results is also given for comparison. The

results are given in calories per gramme molecule. To reduce to foot-pounds per cubic foot multiply by the factor 3·96.

—	800°		1200°		1600°	2000°
—	Clerk	Holborn and Henning	Clerk	Holborn and Henning	Langen	Langen
Air	—	3570	—	5840	8700	11500
CO_2	—	6460	—	10880	17000	23300
H_2O	—	4670	—	7930	14400	19900
Gas-engine Mixture.	4250	3840	6900	6340	9800	13200
Ideal Gas * . .	3430		5400		7350	9300

* $\bar{C} = 4·9$.

The results for the gas-engine mixture are plotted on fig. 126, on which points obtained by Mallard and Le Chatelier's formulæ are also shown.

The energy of gas-engine mixture at 1400° according to Clerk, Holborn and Henning, and Langen respectively would be as follows :—

Clerk	8300
Holborn and Henning	7700
Langen.	8300

It will be seen that the agreement between Clerk and Langen is close, both being about 8 per cent. higher than Holborn and Henning. But it is to be observed that this temperature is just outside the range of all three sets of experiments.

The Committee are of opinion that values of the energy obtained from explosion records are not subject to any very great errors on account of heat loss by conduction to the walls of the vessel, nor on account of incomplete combustion, but that they are affected by errors of quite unknown amount due, first, to heat radiated, and secondly, to the want of thermal equilibrium at the time when the pressure is measured. For the purpose of testing the first of these conclusions, it is very desirable that further experiments should be made on explosions in vessels of greatly different size but of similar form. The opinion entertained by the Committee that incomplete combustion is a surface phenomenon, on which this conclusion as to the validity of the method is based, also requires further confirmation. As regards the second conclusion, further experiment on the actual amount of heat radiated by burning gas is urgently required, and also experiments to confirm or negative the effect of the nature of the wall surface upon the pressure reached in an explosion. The effect of want of thermal equilibrium can be determined up to a point by calculation ; but before such calculation can be usefully made, it is desirable that further information should be obtained as to the temperature distribution after an explosion, especially in the neighbourhood of the walls. It should not be difficult to get an idea of this sufficiently accurate for the purpose by means of platinum thermometers.

The most hopeful way, however, of making use of explosions to give definite information as to the properties of gases would appear to be directly to measure the heat lost in the explosion, as if this be done it is possible to defer the pressure

measurement until such time as equilibrium conditions, except those that depend on the surface of the vessel, have been attained.

The Measurement of Temperature

In all the experiments for the determination of the energy function which

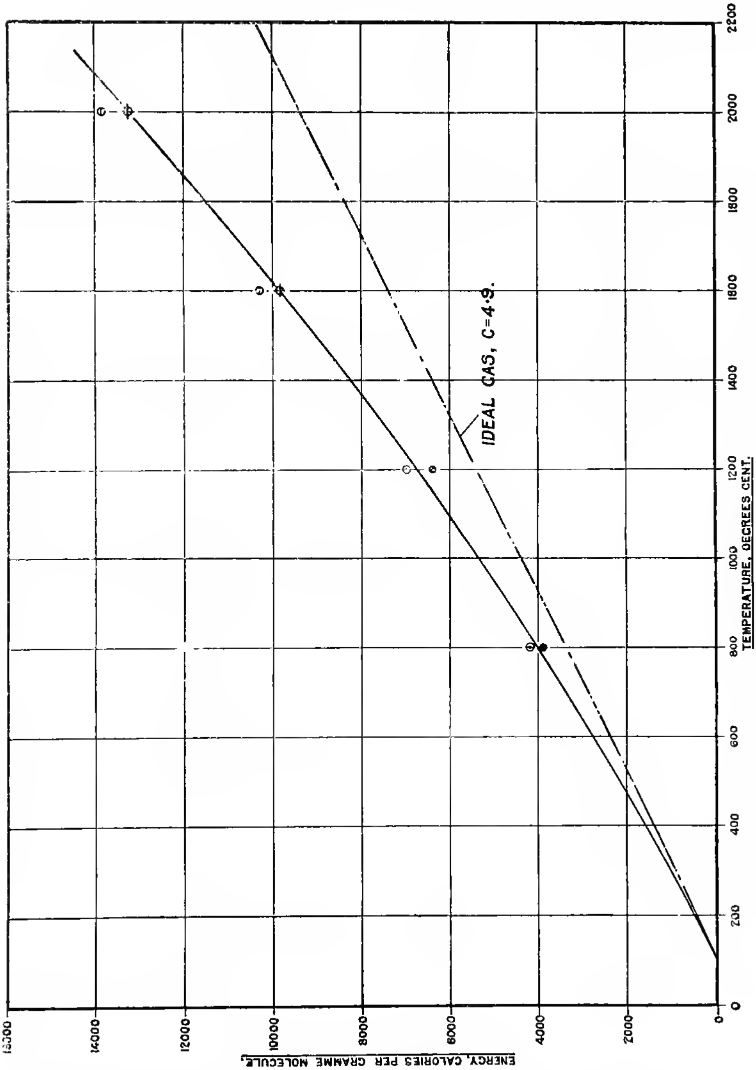

FIG. 126.—Internal Energy of Gas-engine Mixture, composition by volume

N and O	0·83
CO_2 . .	0·05
H_2O . .	0·12
	1·00

Holborn and Henning . . . ●
Clerk . . . . ⊙
Langen . . . . ⊕
Mallard and Le Chatelier . . ⦶

have been described above, the measurement of the temperature is ultimately based upon the pressure or volume changes of the gas. In the constant-pressure experiments of Holborn and Henning the temperature of the gas before entering

the calorimeter was measured by means of a thermo-couple which had been compared with a constant-volume nitrogen thermometer up to 1600° C. In the explosion experiments the mean temperature of a gas is inferred from its pressure. Similarly, in the analysis of the gas-engine diagrams, the gas is itself the thermometer. The mean temperature at any point is taken as proportional to the product pv, and the actual temperature at one point in the cycle (a knowledge of which is necessary for getting absolute values) is obtained either by estimating the quantity of gas present in the cylinder or by direct measurement with the platinum thermometer, as was recently done by Callendar and Dalby.

The temperature scale so obtained is probably sufficiently definite, at any rate for the purpose of gas-engine theory, since the mixture to which it is applied does not vary very greatly in composition, and always consists mainly of nitrogen. It is not so certain, however, that this scale agrees with the absolute thermodynamic scale; and the question of the possible amount of the deviations at temperatures of 1500° and over is of great importance in connection with the present inquiry.

So far as the Committee are aware, the experiments of Joule and Thomson still remain the only comparison between the various gas scales and the thermodynamic scale: this comparison only extended to about 200° C., and is, of course, of no application to the problem now under discussion except in so far as it gives an idea of the differences to be expected at higher temperatures. What it really shows is that thermometers constructed of the more permanent gases are all so closely accordant with the thermodynamic thermometer as to lead to the belief (as a matter of induction, and quite independently of the kinetic or any other theory) that there is really some definite cause tending to make a gas, as such and apart from its composition, obey the law $\frac{pv}{\theta}$ = constant. It would appear that the small deviations from this law, sometimes one way and sometimes the other, which are observed must be due to disturbing causes depending on the nature of the gas, whose influence may be in either direction and is of very various amount, but is at low temperatures small compared with the tendency to obey the perfect gas law. This view being accepted, there is a strong presumption that, if a number of thermometers constructed of different gases be compared at high temperatures and be found to agree fairly well, then they all agree with the thermodynamic scale at least as well as they agree with one another. It is upon the agreement between different gas thermometers that our belief in the measurement of temperature is really founded, and, so far as it goes, the foundation seems to be sound.

The nitrogen thermometer has been used with an iridium bulb up to 1600° C.,[1] but no other gas has been taken above 1100° C. At the latter temperature the differences between thermometers constructed of hydrogen, nitrogen, and air are quite negligible for the present purpose. Rather more deviation has been observed in CO_2; but having regard to the small percentage of this gas which is ordinarily present in the gas-engine mixture, it is not likely that temperatures up to 1100° C., calculated in the usual way from the indicator diagram, will differ much from the true temperatures on the thermodynamic scale. 1100° C. is, however, not much above the lower limit of the gas-engine range; as to what goes on in the upper part of that range we have little or no evidence.

When considerable deviation from the gas laws at high temperatures is observed in the case of any gas, it is usually ascribed to dissociation. For

[1] Holborn and Valentiner, *Ann. d. Phys.* xxii. (1907), p. 1.

example, if comparison be made of two constant-pressure thermometers filled respectively with hydrogen and with iodine vapour they will be found to agree up to about 1000° abs.; the iodine thermometer will then begin to read higher than the hydrogen thermometer until, when the latter reads about 1800° abs., the former will read about double that amount. When the gases are hotter still, the temperature shown on the iodine thermometer will continue to be double that shown on the other. In this case the departure between the two thermometers is accompanied by a change in the absorption spectrum of the iodine vapour; and the whole phenomenon is expressed by saying that the iodine molecule has been split up or dissociated. In the case of a compound gas this dissociation sometimes takes the form of an actual separation of the constituents, which can be detected by diffusion. A great deal of experimental work has been done with the object of ascertaining to what extent the gases CO_2 and steam split up at high temperatures. These gases are constituents in most gas-engine mixtures, and if they dissociate to any considerable extent there will be a corresponding effect upon the $pv\theta$ relations of the mixture of which they form a part. So far, however, there is, in the opinion of the Committee, no conclusive evidence that either steam or CO_2 is dissociated to an extent which is material for the present purpose. Slight traces of dissociation have undoubtedly been found in both cases, but the method of experiment is such as to leave it doubtful how far these have been conditioned by the nature of the walls through which the dissociated gas is diffused. It must be observed, moreover, that CO_2 and steam usually form only a small part of the mixture in the gas engine, and that therefore a considerable amount of dissociation of these gases would be necessary to produce much effect upon the pressure of the whole. Again, such dissociation, if it occurs, must have an effect upon the energy of the gas out of all proportion to the effect which it has upon its temperature. Take, for example, the case of a mixture formed by the explosion of CO and air and containing 10 per cent. of CO_2, the remainder being nitrogen. If, by heating, one tenth part of the CO_2 be split up into CO and oxygen, the resulting change of pressure of the whole mixture will be only one two-hundredth part; but this amount of dissociation could only be effected by the absorption of an amount of heat of the order of 10 per cent. of the total heat of combustion of the gas. In other words, the mean specific heat of the mixture, as determined by the explosion, would be roughly 10 per cent. lower than if there had been no dissociation. Any considerable departure from the gas laws in such a mixture, if it be ascribed to dissociation at all, must therefore be put down to dissociation of the nitrogen, which might conceivably occur at 2000° C., just as iodine vapour is dissociated at a much lower temperature. It does not seem likely, however, that if nitrogen dissociates its splitting up would be accompanied by any visible change in its physical properties, such as is observed in the case of iodine. The phenomenon in this case would be rendered evident only by the departure from the gas law, and possibly by absorption of heat.

It would appear, therefore, that our knowledge of thermometry at these temperatures is more likely to be advanced by direct experiments on the relation between the pressure or volume and the temperature than by looking for other evidences of dissociation. The difficulty in carrying the comparison of different gas thermometers to very high temperatures has hitherto lain in the absence of any material sufficiently refractory to withstand such temperatures and at the same time sufficiently impervious to the gas. Dr. Harker, to whom the Committee are greatly indebted for much information upon this subject,

believes, however, that he is now in possession of a material which will satisfy both of these conditions up to a temperature of 1800° C., and he has suggested that an attempt should be made to compare thermometers constructed with nitrogen, with CO_2, and with argon up to that temperature. If the nitrogen and argon thermometers are found to agree, then, by reason of the great difference in the constitution of these gases, it is almost certain, as explained above, that each agrees with the thermodynamic scale. If, on the other hand, they do not agree, then the presumption is in favour of the argon thermometer, because this gas is supposed to be monatomic and to be incapable of dissociation. The Committee venture to express a hope that a research on these lines will be commenced and carried to a conclusion. They believe that the results obtained will be of very great importance in the investigation of explosions and in the theory of the gas engine, and it seems to them an inquiry eminently fitted for the National Physical Laboratory.

The comparison of gas thermometers is, however, not the only way in which the problem of thermometry at high temperatures may be attacked. Another method, and one that is more satisfactory in some ways because it is more fundamental, is to investigate the dependence of the energy upon the density of the gas. As pointed out at the commencement of this Report, any interdependence between energy and density at a given temperature must be accompanied by a corresponding deviation from the perfect gas law, and investigation of change of energy with density must be the ultimate basis of gas thermometry. The Joule-Thomson experiment was, of course, of this character. Since then Joly has determined the change of specific heat of CO_2 at pressures ranging up to the critical pressure. But these determinations refer only to temperatures of the order of 100° C. As was pointed out at the commencement of the section of this Report dealing with explosions, the corresponding measurement at very high temperatures can be very easily made when once the various corrections necessary to determine internal energy by explosion experiments have been satisfactorily performed. It is only necessary to compare the pressures reached in explosions of mixtures identical in composition but of different density. Should the pressures after explosion, when corrected, be proportional to the pressures before explosion, then the energy is independent of the density, and we have proof that the gas law holds up to the temperature reached by the combustion. On the other hand, a departure from the proportionality would imply a corresponding departure from the gas laws, the amount of which could be calculated. Mallard, Le Chatelier, and Langen have made very careful comparisons of this kind, and they have found that the actual maximum pressures reached in the explosions are in many cases very approximately proportional to the pressures before explosion. Petavel has found that this proportionality is not much altered even when the density of the gas is increased seventy times. This may be regarded to some extent as evidence that there is no very great difference between the gas scale and the thermodynamic scale at the temperatures of 1700°, or more, which were reached in these experiments. But it must be observed that this inference is subject to the same limitations as the determinations of internal energy based upon these experiments. It cannot be regarded as having a secure foundation until the various doubtful questions in regard to heat loss and delayed combustion, which have been raised above in this connection, have been satisfactorily determined.

The Committee think that they can usefully continue their work in the direction of suggesting, and to some extent organising, research on the lines which

have been foreshadowed in this Report. Research of this kind is expensive, and the Committee are of opinion that their work would be greatly facilitated if they had some funds at their command. They therefore recommend that they be reappointed, and ask for a grant of 100*l.*

APPENDIX TO REPORT

By Professor H. L. Callendar, M.A., LL.D., F.R.S., on

The Deviation of Actual Gases from the Ideal State, and on Experimental Errors in the Determination of their Specific Heats

1. The equation $pv = R\theta$, where θ is absolute temperature, is the characteristic equation of a fluid which (1) obeys Boyle's Law at all temperatures, and (2) has the difference of its specific heats constant and equal to R. The specific heat at constant volume or pressure may vary in any manner with temperature, provided that the difference of the two is constant; but both specific heats must be independent of the pressure or density.

For the majority of common gases or vapours (excluding those which polymerise, like sulphur) the deviations from Boyle's Law, as measured by the defect $(R\theta/p - v)$ of the actual volume from the ideal volume, at moderate pressures (say up to ten atmospheres) are to a first approximation a function of the temperature only, and diminish rapidly with rise of temperature. On this assumption, tables of correction for the gas thermometer have been independently calculated by Callendar[1] and D. Berthelot[2] for various gases when employed in the usual manner. The corrections are very small, and agree very closely, though calculated on slightly different assumptions. The differences are much too small to be taken into account in gas-engine experiments.

In dealing with a mixture of gases and vapours at high temperatures, the method of procedure is necessarily somewhat different from the case of the gas thermometer, and the tabulated corrections do not apply. The effective temperature of the mixture is calculated from the value of the product pv/R, assuming that the composition of the mixture is known, and that the constant R has the same value per gramme molecule for each of the constituents as for an ideal gas. The errors involved in this method will be small, and will diminish with rise of temperature, provided that the constituents do not dissociate or polymerise. The experimental evidence at present available with regard to dissociation would indicate that the error of this assumption is certainly less than 1 per cent. for a gas engine mixture at 2000° C., if the composition of the products of combustion is known.

Effective Temperature and Effective Specific Heat

2. Since the temperature of a mass of gas, when exploded in a closed vessel or in the cylinder of a gas engine, is far from uniform, and since the actual distribution of temperature is necessarily somewhat uncertain, it is evident that the variation of the specific heats of the constituents with temperature cannot be *certainly*

[1] *Phil. Mag.*, January 1903. [2] *Trav. et Mém. Bur. Int.*, Paris, 1903.

deduced from a knowledge of the heats of combustion and the effective temperature, even apart from difficulties inseparably connected with the determination of the cooling corrections. It is possible, however, by explosion experiments to deduce values of the *apparent* or *effective* specific heats which, in so far as they approximate to the conditions actually existing in the gas engine, may be of greater practical utility than the true specific heats would be if they could be independently determined. The method of Dugald Clerk, in which the specific heat is directly determined from the work done on the charge after ignition, appears to be particularly appropriate for this purpose.

It is well known that the values of the specific heats deduced from explosion experiments are generally higher than those deduced by more direct methods, and it has been customary to explain the discrepancy largely by possible errors inherent in the explosion method. Such errors undoubtedly exist, and require careful investigation, but in arriving at a decision it is most important to subject other experimental methods to an equally close scrutiny.

Experimental Errors in the Determination of the Specific Heats of Gases by the Constant-pressure Method

3. Apart from errors in the measurement of the temperature of the gas and of the calorimeter, which are not likely to be serious at low temperatures, there is an important source of error in this method, as applied by Regnault and subsequent observers, which has generally been overlooked. In Regnault's experiments, the rate of gain of heat from the heating vessel by the calorimeter was observed before and after the experiment proper, while the gas was not passing through the connecting tube, and was assumed to be the same whether the gas was passing or not. The correction amounted, when the heater was at 200° C., to between 4 per cent. and 5 per cent. of the heat supplied by the gas.

The effect of the gas current would certainly be to change the temperature gradient in the connecting tube in such a manner as to diminish the heat conducted from the heater during the passage of the gas. The error from this cause cannot be exactly determined, but would probably amount to between 2 per cent. and 3 per cent. in Regnault's experiments at 200° C., and would have the effect of making the values as determined by Regnault too low. The true variation of the specific heat of water was unknown in Regnault's time, and he was also unable to correct his thermometers accurately to the absolute scale. These considerations introduce minor uncertainties which might amount to as much as 1 per cent. on the result.

The specific heat of air considered as a mixture of perfect diatomic gases, taking the calorie at 20° C. as equivalent to 4·180 joules, should be 0·2405. Since air is not a perfect gas the actual value must be somewhat greater than this. Regnault's value, 0·2375, is evidently too low.

E. Wiedemann obtained the value 0·2389 by a method similar to Regnault's. This value is probably affected by a similar error.

J. Joly measured the mean specific heat of air at constant volume, and at densities 7 to 22 times normal, by the method of the steam calorimeter, between 10° and 100° C. This method has the advantage of avoiding the majority of the sources of error above mentioned. Joly's value for air at constant volume, when reduced to the calorie at 20° C. and to normal density, would be 0·1732, which corresponds to a value 0·2419 for the specific heat at constant pressure at a temperature of 55° C.[1] This is a far more probable value than Regnault's, but

[1] Callendar, *Phil. Mag.*, January 1903, p. 76.

it must be observed that the extrapolation of the experiments to atmospheric pressure involves some uncertainty.

The specific heats of air and CO_2 at atmospheric pressure and at temperatures of 20° and 100° C. have recently been determined by Swann at the Royal College of Science by the continuous electric method previously employed by Callendar [1] in the case of steam. In this method the actual specific heat at any point is determined by observing the rise of temperature produced in a steady current of gas at the required temperature by supplying a measured quantity of electric energy. This method is better adapted than Regnault's for determining the variation of the specific heat, because it gives the actual specific heat over a small range (about 5°) at the required point in place of the mean specific heat over a large range. It has also the advantage that systematic errors may be more completely eliminated.

The values obtained by Swann for the specific heat of air at atmospheric pressure in terms of the calorie at 20° C. equivalent to 4·180 joules were

$$S = 0{\cdot}2415 \text{ at } 20^\circ \text{ C., and } S = 0{\cdot}2425 \text{ at } 100^\circ \text{ C.}$$

His value at 55° C. is in very good agreement with that deduced above from Joly's experiments by the constant-volume method. Adopting a linear formula, we have for the specific heat at any temperature, t between 0° and 100° C.

$$S_t = 0{\cdot}2413\ (1 + 0{\cdot}00005t) \text{ (Swann).}$$

Holborn and Austin [2] and Holborn and Henning [3] extended Regnault's method for the determination of the mean specific heat to temperatures up to 840° C. In working at these high temperatures the difficulties of the method are greatly increased. They found it necessary to employ electric heating and to connect the heater to the calorimeter by a porcelain tube in order to diminish conduction. The temperature of the hot gas was observed with a thermo-couple near the entrance to the calorimeter. The time of flow was about three minutes in each experiment, and the corrections were estimated by observing the rate of change of temperature of the calorimeter before and after each observation. There appeared to be some doubt whether the couple would give the true mean temperature of the gas flow, especially as the time of flow was so short. For this and other reasons the authors do not lay great stress on the accuracy of the absolute values of the specific heats obtained, but consider that the ratios or relative values, and the rates of increase with temperature, are more likely to be correct than the absolute values, because the various sources of error which they discuss are more likely to be eliminated in the relative values.

The value found for the mean specific heat of air over the range 115° to 270° C. by Holborn and Henning was 0·2315, which is about 5 per cent. smaller than the probable value over this range. For the rate of increase of the mean specific heat they gave the formula :

$$S_{o,t} = S_o(1 + 0{\cdot}00004t) \text{ (Holborn and Austin),}$$

but considered that the rate of increase shown by their experiments was within the limits of probable accuracy of their work, and that it could not be regarded as certainly established that there was any increase over the range of their experiments.

[1] *Proc. R.S.*, 1900. [2] *Sitz. Akad. Wiss.*, Berlin, 1905, p. 175.
[3] *Wied. Ann.* 18, 1905, p. 739.

Later experiments by Holborn and Henning [1] with a platinum heating tube, extending to 1400° C., were made by a similar method, except that the gain of heat by the calorimeter from the heating tube was partly compensated by surrounding the calorimeter at 115° C. with a jacket maintained at a much lower temperature. This compensation was found necessary at high temperatures in order to prevent an excessively rapid rise of temperature of the calorimeter; but although it reduces the apparent magnitude of the correction for gain of heat by the calorimeter, it does not diminish the actual amount of heat transferred and does not reduce the uncertainty of the correction. The magnitude of the effect at high temperatures may be judged from the fact that it was found necessary in the experiments at 1400° C. to maintain the jacket at as low a temperature as 40° C. by passing a stream of cooling water through it in order to prevent the calorimeter rising above 115° C. when no gas was passing. Under such conditions the calorimetric corrections become so uncertain that the probability of systematic errors must increase considerably with rise of temperature. If the method gives a probable error of 5 per cent, in defect over the range 115° to 270° C. it does not seem at all impossible that the error may amount to 10 per cent. over the range 115° to 1400° C.

The rate of increase of the mean specific heat of nitrogen at atmospheric pressure between 840° and 1340° C., shown by the later experiments, was about double that found in the earlier series. Both series of experiments could be represented within the limits of probable error by the linear formula

$$S_{o,t} = 0{\cdot}2350(1 + 0{\cdot}00008t) \text{ (Holborn and Henning).}$$

It appears probable, however, that the value of the specific heat at 0° C. given by the formula is too low, and that the rate of increase is not uniform, but increases with rise of temperature to some extent in the case of nitrogen.

Specific Heat of CO_2

4. Similar remarks apply to the determination of the specific heat of CO_2 by the same methods, but the case of CO_2 is of special interest on account of the rapid variation observed at ordinary temperatures. The following table gives the specific heats of CO_2 according to different observers at 0° and 100° C.:

Temperature	Regnault	Wiedemann	Swann	Holborn
0°	0·1870	0·1952	0·1973	0·2028
100°	0·2145	0·2169	0·2213	0·2161
Increase .	0·0275	0·0217	0·0240	0·0133

The value of the mean specific heat at constant pressure from 10° to 100° C. deduced from Joly's experiments at constant volume is 0·2120, which is nearly 5 per cent. higher than Regnault's value at this temperature, but agrees as closely as can be expected with that found by Swann. The variation of the specific heat with density observed by Joly agrees very closely with that calculated by Callendar [2] from the experiments of Joule and Thomson on the cooling effect in expansion through a porous plug.

[1] *Wied. Ann.* 23, 1907, p. 809. [2] *Phil. Mag.*, January 1903, p. 78.

The rate of increase of the specific heat between 20° and 100° C. observed by Swann is nearly a mean between the rates given by Regnault and Wiedemann, but is much larger than that found by Holborn and Henning, or deduced by Langen from explosion experiments. It is probable that the variation is not linear, but that the rate of increase diminishes with rise of temperature, as indicated by Mallard and Le Chatelier's formula, which would make the specific heat a maximum at 1700° C. The latter formula differs from Holborn and Austin's by more than 20 per cent. at 800° C. The explanation appears to be partly that Regnault's value for the rate of increase at 100° C., adopted by Mallard and Le Chatelier, is too high, but chiefly that Holborn and Austin's values, as already explained in the case of air, are systematically too low, and that the error increases with rise of temperature.

Specific Heat of Steam

5. Regnault's value 0·475 for the specific heat of steam at atmospheric pressure over the range 125° to 225° C. was obtained by taking the difference between the total heats of steam, superheated to these temperatures, as observed by condensing the steam in a calorimeter. Since the difference, corresponding to 100° superheat, is only $\frac{1}{14}$th of the total heat measured in either case, it is evident that the method might give rise to large errors. For this reason many writers have preferred to deduce the specific heat of steam theoretically in various ways from Regnault's value of the rate of change of the total heat of saturated steam—namely, 305 cal. per 1° C.—which, however, really involves the same source of error in an aggravated form. Thus Zeuner gives S = 0·568 ; Perry,[1] S = 0·306 at 0° C. to 0·464 at 210° C. ; Grindley[2] 0·387 at 100° C. to 0·665 at 160° C.

A direct measurement of the specific heat of steam by the continuous electric method gave S = 0·497 at 108° C.[3] Subsidiary experiments, in conjunction with Professor Nicolson,[4] by the throttling calorimeter method enabled the variation of the specific heat with pressure to be calculated. These gave the formula

$$S_t = 0{\cdot}478 + 0{\cdot}0242\, p\, (373/\theta)^{4\cdot3} \quad \text{(Callendar)}$$

where p is the pressure in atmospheres. The approximate constancy of the limiting value 0·478 of the specific heat at zero pressure over the range 0° to 200° C. was verified by calculating the corresponding values of the saturation pressure, which were found to agree accurately with Regnault's observations over the whole range. The theory was also verified by a measurement of the ratio of the specific heats of steam by Makower,[5] which gave values 1·303 to 1·307, agreeing closely with that deduced by Callendar.

The experiments of Lorenz,[6] and of Knoblauch and Jacob and Linde[7] afforded a remarkable verification of the theory of the variation of the specific heat with pressure. They found the specific heat at 1 atmo. to be practically constant over the range 100° to 300° C., but their value, namely, 0·463, is decidedly lower than Regnault's.

Holborn and Henning,[8] in their experiments on the specific heat of steam at atmospheric pressure, improved Regnault's method by employing an oil calorimeter at 110° C. so as to avoid condensing the steam in the calorimeter.

[1] *Steam Engine*, 1899, p. 582.
[2] *Phil. Trans.* 1898.
[3] Callendar, *Proc. R.S.* 1900.
[4] McGill College, 1897.
[5] *Phil. Mag.*, February 1903.
[6] *Forsch. Ver. Deut. Ing.* 21, 1905, p. 93.
[7] *Loc. cit.* pp. 1 and 35, 1906, p. 109.
[8] *Ann. Phys*, xviii. 1905, p. 739.

They determined the ratio of the specific heat of steam to that of air by passing currents of air and steam in succession through the apparatus under similar conditions, and obtained the following values of the ratio for different intervals of temperature :

Temperature Interval .	110°–270°	110°–440°	110°–620°	110°–820°
Ratio/Steam : Air .	1·940	1·958	1·946	1·998

In their subsequent series with a platinum heating-tube at higher temperatures they obtained the following ratios :

Temperature Interval .	115°–826°	115°–1180°	115°–1324°
Ratio/Steam : Air .	1·900	1·973	2·003

The second series appears to make the ratio about 5 per cent. lower at 110°–820° than the first, which suggests the possibility of constant errors depending on the type of apparatus employed or on the velocity of the gas-current. The experiments of Callendar and Swann would make the ratio 2·05 at 100° C., which is higher than any of the values obtained by Holborn and Henning at 1400° C.

Holborn and Henning point out that their results at 1400° C. cannot be reconciled in the case of steam and CO_2 with any of the results of explosion methods. They are 6 per cent. to 13 per cent. lower than Langen's, which are among the lowest. But having regard to the fact that the constant-pressure method which they employed appears to give results so much lower than Joly's or Callendar's methods at ordinary temperatures, and that the experimental difficulties increase so greatly at higher temperatures, it does not seem at all improbable that a considerable part of the discrepancy is to be attributed to systematic errors of the constant-pressure method.

On the Cause of the Variation of Specific Heat

6. It appears from theory that the energy of translation and rotation of the molecules of an ideal gas should vary in direct proportion to the product pv. The internal energy of vibration of the molecules, however, which is related to the absorption or emission of radiation must vary by the Stokes-Kirchhoff law in relation to the full radiation of a black body at the same temperature. According to Planck's formula, which has been verified over a very wide range, the energy of full radiation corresponding to wave-length L in full radiation, varies with the temperature according to the expression

$$E_L = CL^{-5}(e^{c/L\theta} - 1)^{-1}$$

the value of the constant c is 14,700 if L is measured in microns, μ, or millionths of a meter. The energy of vibration of a molecule which is in equilibrium with full radiation at any temperature will depend on the extent to which its free periods of vibration respond, as indicated qualitatively by its absorption spectrum. Those periods which respond very strongly may produce an appreciable effect on the specific heat.

It happens, for instance, that CO_2 has a very marked absorption band at 15μ, nearly, which can be detected even when the gas is present in small quantities in the atmosphere. So far as this particular mode of vibration is concerned, the specific heat would increase most rapidly at ordinary temperatures, which is actually observed to be the case with CO_2. According to Planck's formula, the effect of any mode of vibration would be a maximum when θ is infinite, and

would then contribute the term C/cL^4 to the mean specific heat ; but for $L = 15\mu$ the effect would have already reached within about 10 per cent. of the possible maximum at 2000°. According to Wien's original formula

$$E_L = CL^{-5}e^{-c/L\theta}$$

which holds very well for short wave-lengths and low temperatures, but appears to fail when $L\theta$ is large, the energy E would reach a finite limit CL^{-5} when θ is infinite, and the specific heat for $L = 15\mu$ would reach a maximum when $\theta = 500°$. This does not appear to agree so well with the changes of specific heat actually observed.

In the case of steam it appears that there are no equally well-marked absorption bands corresponding to strong natural periods of vibration, in the range of the heat spectrum available for investigation. The very high dielectric constant of water for short electric waves has been taken to indicate that there is a period of marked resonance very low down in the spectrum in the unexplored field between the shortest electric waves and the longest heat waves hitherto obtainable. This might account for the relatively high value of the specific heat of water and steam at ordinary temperatures. It must be remembered, however, that the absorption spectrum is very complicated, and difficult to investigate beyond the limits of photography. Moreover, it is very difficult to deduce, except in a qualitative manner, the relative intensities of the energy corresponding to each absorption band. An absorption band may appear strongly marked in a thick layer of absorbent, which really corresponds to a very small amount of energy. For this reason no quantitative estimation of the effects of vibration of the molecules on the specific heat is possible at the present stage of knowledge, but it is important to bear the possibility of such effects in mind as a guide for future investigation.

APPENDIX V

Adiabatic and Isothermal Compression of Dry Air

(*Professor R. H. Thurston, Journal of Franklin Institute*, 1884)

One hundred volumes of dry air at the atmospheric mean temperature of 15°·5 C. and 14·7 lb. per square inch undergo change of volume without loss or gain of heat. The temperatures and volumes corresponding to various pressures are given. Also the volumes at the various pressures if the temperature remained constant at 15°·5 C.

Absolute pressure in lbs. per sq. in.	Temperature of compression in Centigrade degrees	Volume at temperature and pressures preceding	Volume if temperature constant at 15°·5 C.
14·7	15·5	100·0	
15·0	17·26	98·58	98·00
20·0	42·60	80·36	73·50
25·0	64·76	68·59	58·80
30·0	82·10	60·27	49·00
35·0	98·38	54·01	42·00
40·0	113·86	49·13	36·75
45·0	126·54	45·18	32·67
50·0	138·96	41·93	29·40
55·0	150·53	39·19	26·73
60·0	161·38	36·84	24·50
65·0	171·61	34·80	22·62
70·0	181·29	33·02	21·00
75·0	190·49	31·44	19·60
80·0	199·26	30·03	18·38
85·0	207·66	28·77	17·29
90·0	214·71	27·62	16·33
95·0	223·45	26·58	15·47
100·0	230·91	25·63	14·70
125·0	264·66	21·88	11·76
150·0	293·91	19·22	9·80
175·0	319·87	17·23	8·40
200·0	343·31	15·67	7·35
225·0	364·71	14·41	6·53
250·0	411·57	13·38	5·88
300·0	420·34	11·75	4·90
400·0	480·76	9·58	3·90
500·0	531·21	8·17	2·94
600·0	574·93	7·18	2·45
700·0	603·74	6·44	2·10
800·0	648·80	5·86	1·84
900·0	680·86	5·39	1·63
1000	710·49	5·00	1·47
2000	929·67	3·06	0·74

INDEX

TO

THERMODYNAMICS

OF THE

GAS, PETROL, AND OIL ENGINE